COMBUSTION

EVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

February 1954

Coal unloading dock and storage facilities of Electro Metallurgical Co., Marietta, Ohio

Remote Control Features Coal Handling

Sulfuric Acid Influence On Dewpoint

Measurement Errors In Plant Tests

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

Vol. 25 **No. 8**

February 1954

Feature Articles

Editorials

Departments

COMBUSTION publishes its annual index in the June issue and is indexed regularly by Engineering Index, Inc.

GERALD S. CARRICK
Business Manager

JOSEPH C. McCABE
Editor

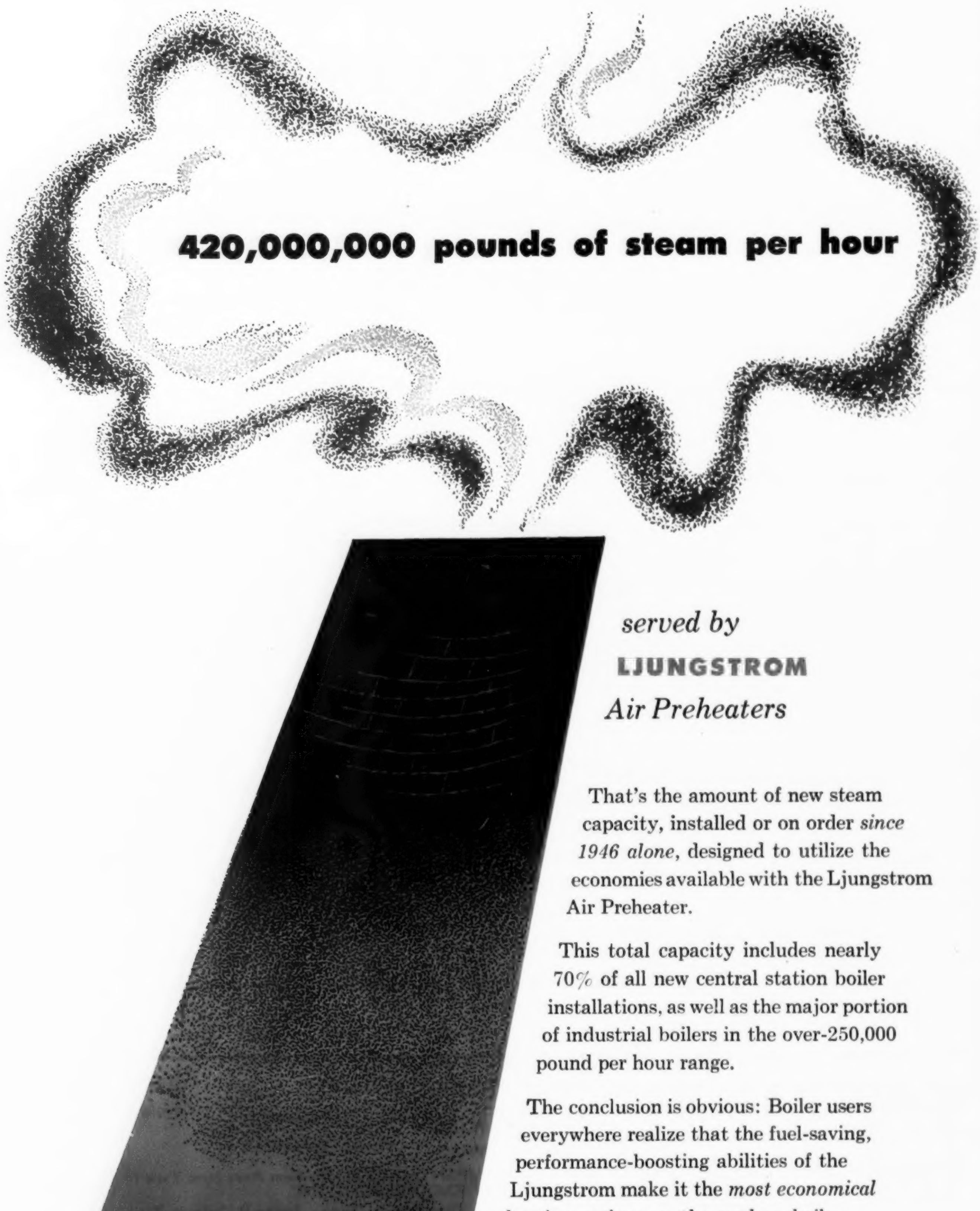
GLENN R. FRYLING
Associate Editor

CARL G. UDELL
Circulation Manager

Published monthly by COMBUSTION PUBLISHING COMPANY, INC., 200 Madison Ave., New York 16
A SUBSIDIARY OF COMBUSTION ENGINEERING, INC.

Joseph V. Santry, Pres.; Charles McDonough, Vice-Pres.; Otto Strauss, Treas.; Irving B. Swigart, Secy.
COMBUSTION is sent gratis to engineers in the U. S. A. in charge of steam plants from 500 rated boiler horsepower up and to consulting engineers in this field. To others the subscription rate, including postage, is $4 in the United States, $4.50 in Canada and Latin America and $5 in other countries. Single copies: Domestic, 40 cents; Foreign, 40 cents plus postage. Publication Office, Easton, Pa. Issued the middle of the month of publication.
Acceptance under Section 34.64, P. L. & R., authorized by United States Post Office.

Printed in U. S. A.


420,000,000 pounds of steam per hour

COMBUSTION

Editorials

Railroad Electrification

At a time when increase of generating capacity in central stations is continuing to record heights it may be well to inquire about the status of one of the early major power users, the electrified railroad. This was done by H. F. Brown and R. L. Kimball of Gibbs & Hill, Inc. who presented a paper at the AIEE Winter General Meeting entitled "A Re-Appraisal of the Economics of Railway Electrification—How, When and Where Can It Compete With the Diesel-Electric Locomotive?"

In the United States most of the major electrification projects were placed in service between 1905 and 1925. Since 1938 the number of electric locomotives in service has been constant at about 800. No important new projects have been undertaken in this country since that time, although there have been quite a number completed abroad.

The problem is primarily one of economics, and particularly of justifying the large initial investment required for electrification. At the present time it is estimated that the heat rate of a diesel-electric locomotive is about 12,400 Btu per kwhr. Operating cost for this mobile power plant is nine mills per kwhr, assuming diesel oil at ten cents per gallon. Messrs. Brown and Kimball pointed out that this is already higher than the energy costs under some existing railroad electrification power contracts. Means must be found for attracting the capital required for a wholesale change to electrification, as contrasted to the gradual substitution of motive power unit by unit.

Perhaps the most favorable aspects for electrification are these. Power is available from private sources, so that it will not be necessary for railroads to generate their own, as was the case in the early installations. The technology of electrification is quite thoroughly understood and is capable of meeting foreseeable demands for extension of electrified railroads.

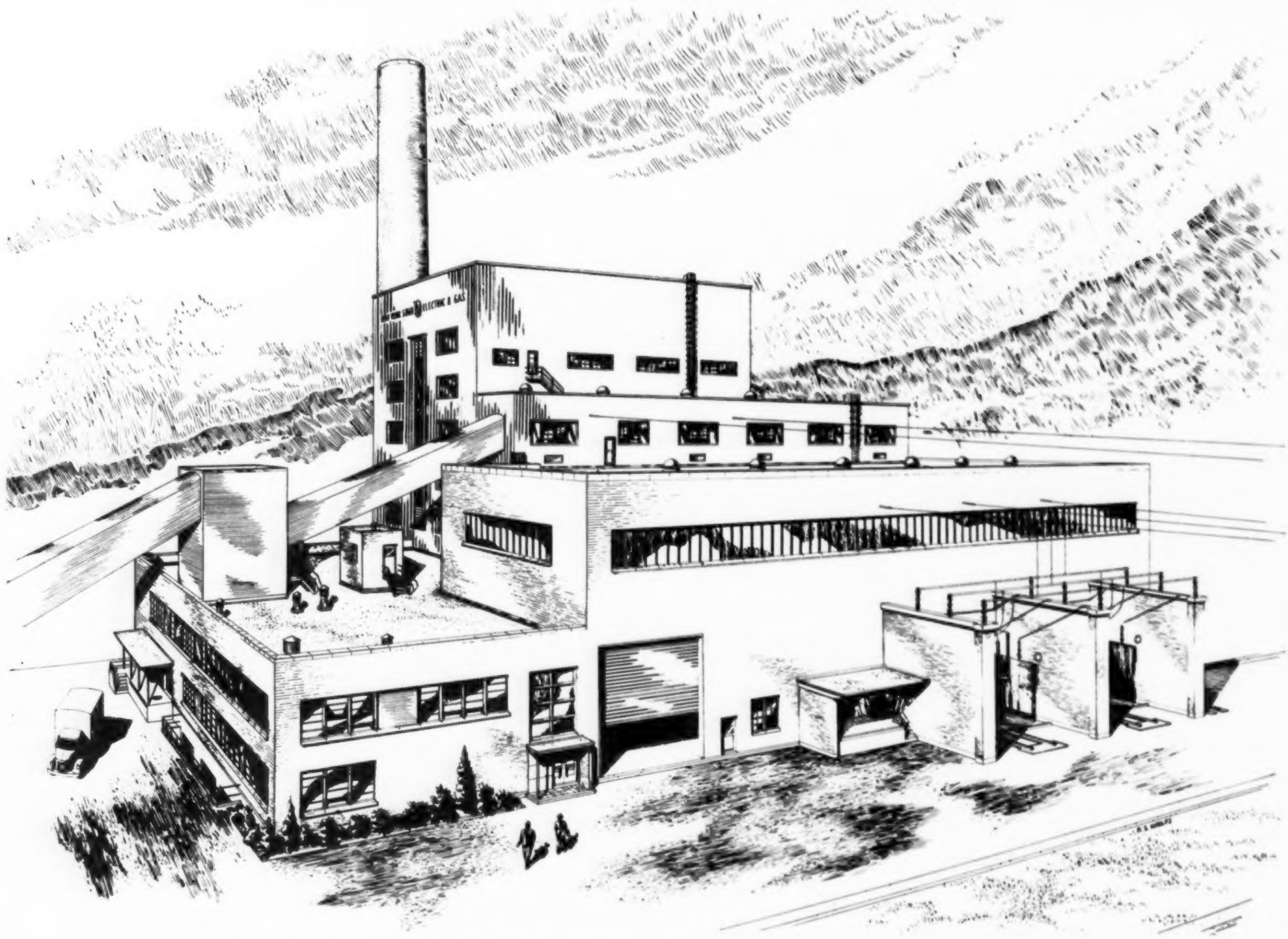
The ultimate deciding factors will likely center upon load-building programs of public utilities and the willingness of railroads to make the necessary investment for any sizable, long range program.

Research and Supply and Demand

The very recently completed Winter General Meeting of the AIEE (Jan. 18–22, 1954) and the last Annual Meeting of the ASME devoted a healthy share of their programs to papers discussing specific basic research projects or to contemplating certain directions for research to take in fields of prime concern to their membership. One of our contemporary publications pushed the interest a little farther and ran an editorial suggesting active participation for one major class of power equipment user in product development.

All of the above, to our mind, are merely present-day manifestations of the national temperament. As a people we have been blessed with a heavy measure of impatience. One that in earlier days had its outlet in the form of Yankee ingenuity. So to see this eagerness to get in and speed things up by self participation still very much alive is an encouraging sign of vigor. As a matter of fact much of our present-day power technology has resulted from a close relationship between the consuming industries and the prime suppliers. Often the consumer has demanded materials or designs to raise overall efficiencies beyond anything known at that time and the suppliers have met the demand.

Today with the power industry on the threshold of higher pressures and temperatures and basically new methods of power generation, the capital costs involved for new product development will run extremely high, so much so that the brand new materials capable of performing at these high levels will have to undergo long, careful fundamental research before the application stage. But because the demand for their development is strong we feel certain the proper materials will be forthcoming and probably from their traditional sources, the ones most keenly attuned supply and demand.



Milliken Station of New York State Electric & Gas Corp. has elected to break with the traditional coal-handling practice of selecting equipment sized for intermittent service in favor of smaller, continuous automatic coal moving devices

Continuous Coal-Handling System Employs Outdoor Live Storage Pile

By making an outdoor coal pile part and parcel of its live storage and then controlling coal movement from it by remotely-operated automatic devices, Milliken Station of New York State Gas & Electric Co. stands to reap the benefits of lower installation, maintenance and operating costs.

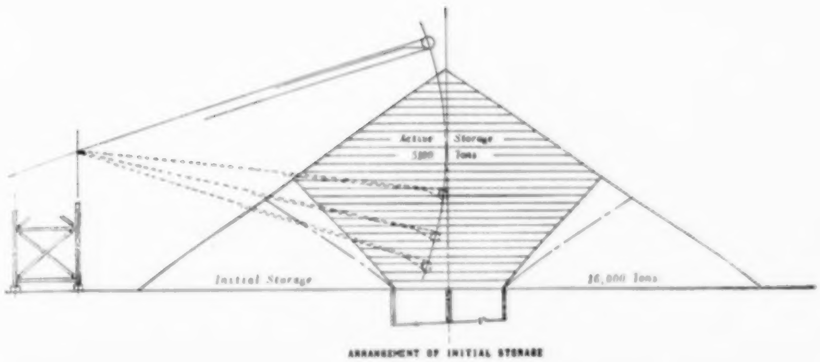
by H. C. SCHWEIKART
Gilbert Associates, Inc.,
Reading, Penna.

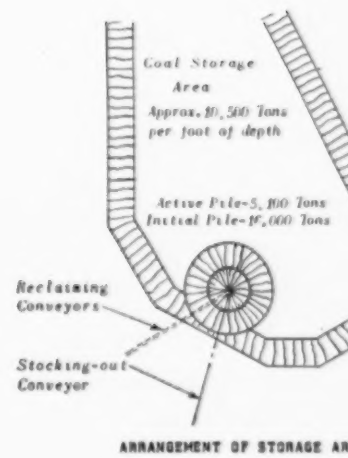
COAL-handling facilities in modern power plants cost from \$5.00 to \$20.00 per installed kw and average out at about 7 per cent of the power plant cost. Unlike most of the other major elements such as the turbine generators, steam generators, and associated auxiliaries, coal handling has not made its share of the last decade's progress in lowering installation or maintenance costs. For example, coal-handling system design still follows the principle of furnishing enough equipment to permit intermittent, manually-supervised, 40-hr week operation.

The new Milliken Station, above, of the New York State Electric & Gas Corp. breaks with this pattern. It employs a wide use of remotely-operated automatic controls combined with a systematic use of an outdoor coal pile as part and parcel of live storage. With reliable, carefully controlled automatic aids a continuous coal feed flows from this outdoor pile, Figs. 1–3, into an indoor storage bunker at a preselected rate based on burning. This bunker serves primarily for emergency needs.

Handling System Design

The preliminary studies of coal-handling facilities at Milliken followed the conventional pattern. But soon it appeared that a continuous coal flow to the pulverizers using conventional equipment and holding reserve storage to a minimum would eliminate the hazards and high investment costs of large bunkers. Further it promised





1—Separate outdoor coal pile, above, built up by a stacking-out conveyor, feeds a continuous stream of coal to in-plant bunkers or, if so desired, incoming coal goes to conventional 500,000-ton storage pile, drawing, right, for later reclaiming by way of the active pile above

to reduce operating labor costs as well.

Investigation of the work of others plus the ideas of the utility company engineers and ourselves led to the continuous system, Figs. 1–3, that meets plant needs in every particular and at a lower initial investment cost than for a conventional design. Mr. Walter Greacen, Chief Mechanical Engineer for New York State Electric & Gas Corp., pointed out in a prepared discussion of our paper[1] before the ASME that the ultimate investment savings by the time Milliken Station's fourth unit goes in will reach about $600,000 and will return, in the meantime, $6000 to $7000 annually in operating labor savings.

At Milliken, coal arrives, Fig. 2, in railroad cars on sidings large enough to accommodate 85 cars. Then a 50-ton diesel-electric, company-owned locomotive hauls the loaded cars over a series of individual thawing pits to a car dumper where they are weighed and dumped into a car hopper feeding the plant system. Empty cars are shifted to a 60-car capacity railroad siding for pickup by the railroad company.

The individual thawing pits, 17 in number, are especially arranged so a group of three coupled cars of any size can be thawed together. When the middle car of the three is over the center pit each individual hopper, regardless of car length or number of hoppers, stands over a thawing pit. The center car covers a set of five pits spread over a length of 26 feet. Both the other cars reach across six pits measuring 31 feet in length. Each set of pits is 15 feet apart.

Two conventional burners equipped with oil control and pilot bypass valves are in each of the pits. At the beginning of the work period one burner in each pit is put on low fire and the other is left off. Air is admitted, though, to each burner under low pressure through the minimum restriction holes in the butterfly valve. This air is just enough to keep the burner nozzle cool in the idle burner and produce a rather lazy flame of about a foot in length on the working burner.

Once a group of three cars are positioned over the pits the attendant selects which pits he'll operate by opening the appropriate valves on the panel board. The burners in these pits then come on at full rating. Those thawing pits not under hoppers remain with one burner running at the pilot setting. As soon as the cars have thawed enough the working burners are cut back to their former settings and the cars are moved onto the dumper.

The car dumper, a 90-ton rotary design, has an operating cycle of 90 seconds. It takes, though, an average of 110 seconds to weigh, unload and shift the individual cars since they are not all of the same size. After coal is dumped it falls into a car hopper which has two outlets. The outlets each have a 450-tph frozen coal cracker and an apron feeder. These feeders discharge onto a single 900-tph belt conveyor, called the stocking conveyor, that incorporates a hinged boom at the yard storage area discharge end. (See Coal Storage Details, below.) In this way all coal reaching the station goes first to storage and is controlled manually on an intermittent schedule.

Yard storage consists of an inactive, conventionally stored coal volume of about 500,000 tons and an active volume of 5100 tons, Figs. 1, 4. This active, outdoor storage pile is not a new idea in itself. R. P. Moore, mechanical design engineer, Niagara Mohawk Corp., has reported one installation at his company's Huntley No. 2 Station, Fig. 5, which was built to connect outdoor bunker storage for direct feed to coal pulverizers. F. H. Schiffer of Public Service (N.J.) Electric & Gas Co. and T. S. Fetter, Jr., mechanical engineering division, Philadelphia Electric Co. also have recalled the use of outdoor active storage piles in a number of stations on various utility systems. What makes the Milliken Station design different is the first application of an outdoor active pile fed to a plant at a preselected coal-burning rate under the remote control of belt weighing devices. Here is how the system works.

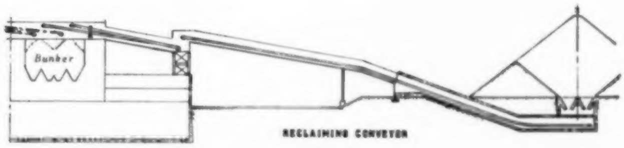
Live Storage Operation

The active 5100 ton storage is built over four reclaim hoppers, each having 300-tph remote-controlled feeders. Each pair of feeders delivers coal to 300-tph crushers.

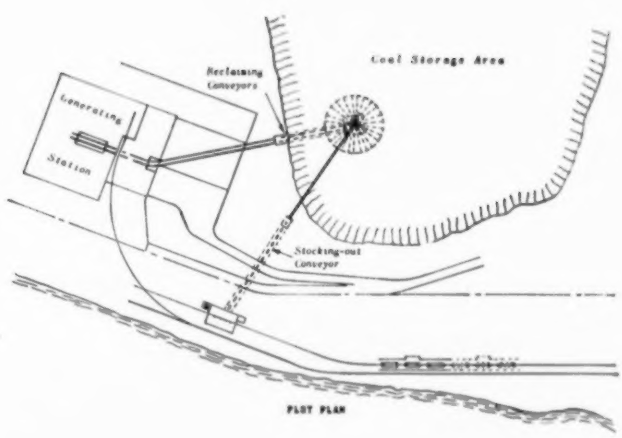
Coal discharges from these crushers to either of two 300-tph belt conveyors equipped with weighing devices to control the weight of the coal on the conveyor. The two belts elevate the coal to a transfer point at the power plant. Here the coal discharges onto two similar belts feeding coal to the first of an eventual series of bunkers. There is no tripper, shuttle conveyor or similar device. The two belts, while similar, are of different lengths so the coal discharge can be staggered the length of the bunker. This portion of the coal handling is intended for continuous operation at the rate set by operators in

[1] "Coal Handling Facilities for Milliken Station With Automatic Remote Controls" by H. C. Schweikart, Gilbert Associates, ASME Paper No. 53-A-28.

2—Coal, arriving by railroad, passes over thawing pits, to a rotary car dumper, onto a stacking-out conveyor



3—Active storage pile feeds four reclaim hoppers connecting to a pair of 300-tph crushers serving two reclaiming conveyors



4—Overall layout of coal-handling facilities pictures the relationship of the different major working elements

the control room. The last bunker in operation will fluctuate somewhat in level and require observation of the operator to maintain a proper adjustment of feed rate to burning rate.

The chute leading from the discharge end of these belt conveyors has a flanged slot to permit inserting a cut-off plate to stop all coal flow when desired. A further opening has been built into the chute for feeding later coal bunkers. As the plan stands, this opening, now closed off, will feed a secondary chute leading from the opening to a future 300-tph belt conveyor serving this later bunker. Then coal will discharge to the first or present bunker until it fills to a height where coal in the chute rises above the slot feed to the second bunker's chute.

Bunker Selection

The final individual bunker capacity, 700 tons, for the first part of the plant was reached by setting its dimensions to conform to the power plant building lines. Its height was established by extending the roof above the bunker at the same elevation as the turbine room roof. Its length and width were fixed by the space required for the pulverizing equipment. The 700-ton capacity is sufficient to operate the steam generator for 14 hrs at a maximum rating of 1,000,000 lbs per hr and a firing rate of 50 tons per hr.

It is our considered opinion that the bunker could have been still smaller, of the order of 400 tons or 8 hrs operation, with this continuous automatic reclaim scheme and still not jeopardize continuity. In fact we feel the present 700-ton bunker provides more active storage than a 2000-ton bunker gives in a conventional installation.

Each bunker carries four bin indicators, one above each of the four pulverizers. They cover a range of depth of several feet with remote indications on the panel board in the control room. Bunker tops are completely closed.

Twin gates are set in the bottom of the bunker. These two gates are housed in a single frame having in effect two inlets and one outlet. A steel division plate between the gates extends upwards into the bunker just above the point where the bunker wall becomes vertical. Experience shows that a bunker having at least one straight side gives better results than one where all sides slope regardless of the angle.

Coal leaving the bunker by way of the gates passes through a chute to an automatic coal scale having a weight hopper capacity of 400 lbs. Fig. 6 shows the arrangement of gates, scale and feeders, and chutes between bunker and pulverizing mill.

To aid in maintaining coal flow from bunker outlet to mill inlet as near a vertical drop as possible was provided. All chute hoppers are sized by the smaller of the twin gate openings to maintain active continuous movement throughout. Again the idea of a straight side is followed where possible.

Steam Generators

There are four mills for each steam generator, all on the basement floor. Each mill receives its coal from a separate feeder, Fig. 6, controlled by a vari-stroke, variable speed drive. The mills have a normal pulverizing capacity of 33,500 lbs of coal per hr based on a coal of 57 Hardgrove grindability, a maximum moisture of 5 per cent and pulverizing to a fineness of 70 per cent through a 200-mesh screen. This capacity permits three mills to supply full steam generator rating needs when burning average quality coals. The four mills have ample capacity for full steam generator rating when burning the poorest quality of coal expected.

No provision has been made to burn any other fuel except light fuel oil on an emergency basis such as a prolonged rail or coal strike.

Coal Storage Details

The role of the belt conveyors, Figs. 2, 3, in the Milliken Station coal-handling system is vital. Because of this some special details on their selection and operation are well worth while. As we pointed out earlier under Handling System Design there are two apron feeders receiving the output from the rotary car dumper by way of individual frozen coal crackers or crushers to reduce all lumps to 6 in. or less. These apron feeders discharge the coal to a single 900-tph belt conveyor that has a 147.5-ft lift on 480-ft centers.

The last 113 feet of this belt conveyor, known as its discharge end, is designed as a lowering boom to reduce the dust nuisance. The boom is raised automatically as the initial pile of coal changes in height up to the design limit of 70 feet above the reclaim hoppers.

When the coal reaches this 70-ft height it has a stocking

out angle of repose of 36 degrees. The base of the conical pile at this 70-ft height, Fig. 1, measures 190 ft in diameter. The pile contains 16,000 tons. The four reclaim hoppers roughly at the center of the pile's base present an opening 33 ft square. With a reclaim flow angle of 50 degrees the active quantity of coal available for automatic reclamation runs about 5100 tons.

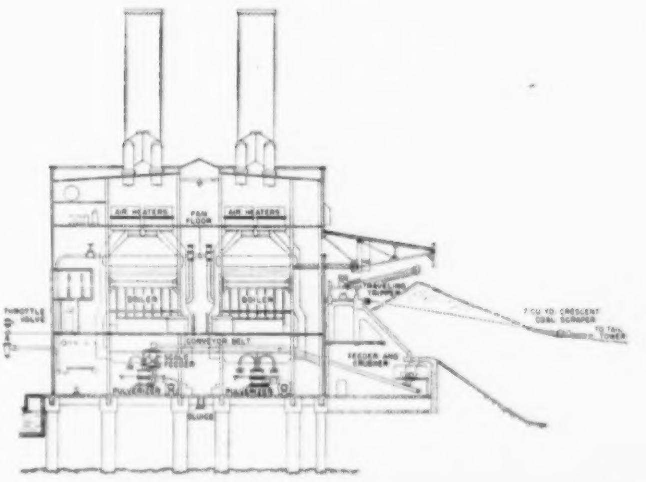
This 5100 tons added to the 700 tons in the bunker gives a combined active storage total of 5800 tons, enough for 116 hrs of operation at the maximum burning rate of 50 tph. When a second steam generator goes in, also with a 700-ton bunker, the total tonnage ready for automatic reclamation will be 6500 tons, enough for 65 hrs operation at the then maximum burning rate of 100 tph. It is obvious, though, that as more steaming units with their 700-ton bunkers go in the hours of available coal supplies drop.

In actual operation as additional units go in, however, it is unlikely that full load will be carried on all units at the same time, particularly at night and on week ends. Considerable reduction of total station generation can be expected, perhaps of the order of 50 per cent station load factor, over an extended week end involving holidays on Friday or Monday. But even with exceptionally high total station generation there would probably be enough fuel to last 62 hours or so. If it should be necessary to reclaim coal from the non-active storage area a bulldozer and operator on a single shift per day would be ample.

Incidentally we considered other possibilities for dustless stocking out of coals beside the hinged boom we selected. One was a 70-ft high, 10-ft diameter unloading tower. But its costs, some $20,000 over the hinged boom plus more for foundations, ruled it out. Further we felt the operation and maintenance of the tower's hinged doors, the power requirements and the obstruction in the center of the coal pile outweighed any expected advantages.

Telescopic chutes were another possibility. Their first cost looked about $10,000 cheaper than the hinged boom. But against this advantage was our fear that they (1) might not be as effective in the control of dust, (2) might prove troublesome in high winds and during freezing rain or sleet, (3) would require more power.

Some consideration was also given to use of a short telescopic chute or short unloading tower in conjunction with the hinged boom. Either would add $10,000 to $12,000 to coal-handling costs and at no appreciable advantage.



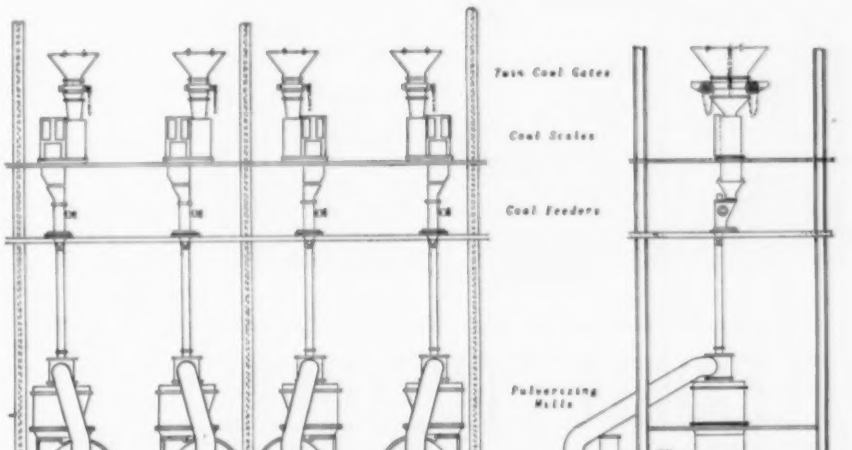
5—Niagara Mohawk Corp. installed the above system at its Huntley No. 2 Station to connect outdoor bunker storage for direct feed to coal pulverizers within the plant

Reclaim Layout

The reclaim structure and its four individual hoppers, all of concrete, has steam heating coils imbedded in the concrete hopper to prevent any freezing. A combination control and cut-off gate handles the coal discharge to vibrating feeders. These feeders provided with motor driven rheostats for automatic control by continuous weighing devices have, in conjunction with the gates, a range in capacity from zero to 300 tph. The gates, though, can be set manually for extreme variations in flow.

Two of the feeders discharge into a single, ring type double roll crusher while the other two feeders discharge into a second similar crusher. Each crusher has a 300-tph capacity when delivering a finished product that can pass through a 1-inch ring. High-speed crushers were picked as a solution to the plugging nuisance presented by coals with high percentages of clay and other impurities. The crusher discharge chutes have flap gates so either

6—Indoor, 700-ton bunkers, have twin gates set in the bottom to feed over scales onto feeders and then to pulverizing mills. A steel division plate between the gates extends up into the bunker to prevent arching or ratholes



crusher can discharge coal to either belt but not both simultaneously.

The two 300-tph, 375-fpm, 30-in. wide belts receiving coal from the crushers each have a continuous weighing device to maintain a preselected total coal flow. This flow rate is based on output from two of the four feeders regardless of the distribution through each feeder. It is remotely set by the operator guided by a dial and calibrated scale in the control room to as near one-half the actual burning rate for all units in service as possible.

Both belts and crushers and all four feeders should run at reduced rating as much of the time as possible to get maximum use of the active storage pile and the bunker. Normally all coal reclaimed from storage will be divided on both belts at the same time. But the operation is highly flexible. Any one scale in the control room can control any combination of feeders from one to four and deliver all coal on one belt which may be desirable to reduce station power with only one or two units installed.

The two rate setting instruments, and two remote tonnage indicators, one for each belt, will be located on a panel board in the control room. These instruments will be provided with a dial and/or a scale calibrated in tons per hour from 0 to 300 tph for remote control setting of the coal being conveyed.

Metals and Materials

Because specifications for composition and working stresses of high-temperature materials differ in various countries and the published test results for creep of similar type steels show substantial differences. **W. E. Bordgett,** research manager, United Steel Co., Rotherham, England, and **Dr. C. L. Clark,** metallurgical engineer, Timken Roller Bearing Co., presented a paper "Comparative High-Temperature Properties of British and American Steels," at the 1953 ASME Annual Meeting.

Five representative high-temperature steels in general use in both countries were chosen for testing in the authors' individual company laboratories. Three were ferritic steels, two austenitic. Stress, temperature and time adopted were the same but no attempt was made at a standardized test procedure. The creep tests showed excellent reproducibility between the two laboratories for ferritic steels but substantial differences for the austenitic steels. The authors advanced several reasons for these differences.

In general, though, the limited tests did indicate that for certain steels there is no justification for differences in stress specifications in the two countries. The austenitic steels with their composition and grain structure differences pointed up the need for further study of these factors to achieve the highest standard of creep resistance.

A companion paper "A Critical Examination of Procedures Used in Britain and the United States to Determine Creep Stresses for the Design of Power Plants for Long Life at High Temperatures," by **Dr. R. W. Bailey,** consulting research engineer, Metropolitan-Vickers Electrical Co., Ltd., examined the more commonly used test procedures in each country.

Dr. Bailey took a very strong stand in expressing his disappointment with the present testing procedures in both countries. These methods of making tests at the working temperatures, and then extrapolating on a stress basis without regard to the effect of thermal action, Dr. Bailey felt, constituted a mistake in the particular case investigated and produced unreliability and uncertainty regarding design-stress values where materials and their behavior for long life at high temperatures are concerned. He, instead, advocated a method of adopting a working stress as the test stress and of using temperature to accelerate creep as the only means of judging correctly the influence of particular elements in relation to their life period and the working temperature to be borne.

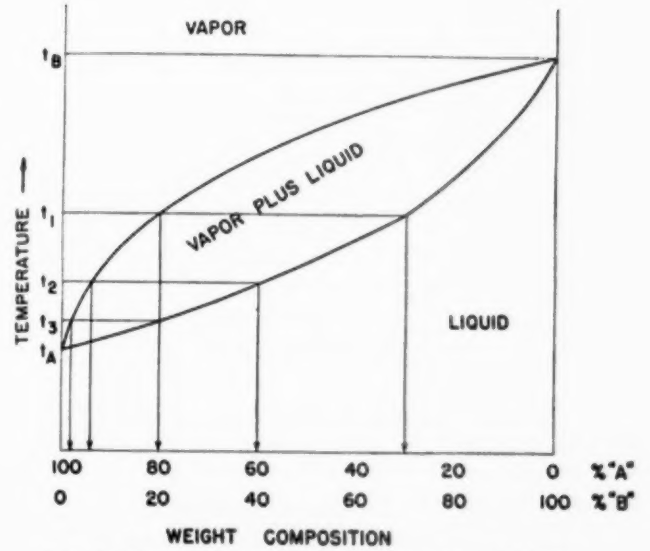


Fig. 1—Representation of vapor-liquid equilibrium

TABLE 1—SULFURIC ACID-WATER BOILING POINT DATA
(Applicable to Fig. 2)

Acid Concentration, Per Cent by Weight	Slope, m	$t_w = mt_a + b$ Constant, b
0	1.000	0
10	0.976	0
20	0.972	−1
30	0.937	−3
40	0.906	−6
50	0.889	−18
60	0.884	−41
70	0.875	−74
80	0.822	−112
90	0.748	−157
95	0.670	−174
98.3	0.639	−197

TABLE 2—CONVERSION FACTORS

Constituent	H_2SO_4	H_2O
Weight per cent	X	$100 - X$
Weight fraction	$\frac{X}{100}$	$\frac{100 - X}{100}$
Mols	$\frac{X}{9800}$	$\frac{100 - X}{1800}$
Mol fraction	$\frac{9}{4900/X - 40}$	$\frac{4900/X - 49}{4900/X - 40}$
Mol per cent	$\frac{90}{490/X - 4}$	$\frac{49000/X - 490}{490/X - 4}$

The Influence of Sulfuric Acid Upon the Dew Point of Combustion Gases*

In this article which originally appeared in the November 1953 issue of the *Journal* of the American Society of Naval Engineers, Inc., the authors show how the highest dew point of the combustion gases from sulfur bearing fuels may be predicted. The method is developed from a consideration of basic equilibrium data.

By GEORGE A. FEARN, JR.

and

COMMANDER WILLIAM TESSIN, USN

THE behavior of sulfuric acid vapors in the flue gases may be illustrated as in Fig. 1. The upper curve shows the condensation temperature as a function of the vapor composition and the lower curve shows the liquid boiling point as a function of composition. The diagram is applicable to a single total pressure; similar diagrams apply at other pressures. For a pure component, the boiling point and condensing temperature are the same. The boiling point of constituent B is t_B and that of A is t_A.

To illustrate the use of such a diagram, assume a vapor of composition 20 per cent B, 80 per cent A. The first condensate appears at t_1, the dew point; the composition of the condensate is rich in component B; namely 70 per cent B, 30 per cent A. At a temperature as t_2 the vapor composition has changed to 40 per cent B, 60 per cent A. At a temperature as t_3, the last vapor present, has a composition 2 per cent B, 98 per cent A, while the liquid has the original composition of the vapor; namely 20 per cent B, 80 per cent A; that of the condensate changes from 70 per cent B, 30 per cent A to 20 per cent B, 80 per cent A.

The above discussions may be extended when inert gases are mixed with the vapors. However, when cooling below a temperature as t_3, some of the condensables will remain in the vapor state to saturate the inert gases. The liquid composition will then change as the partial pressure of the vapor changes.

To draw a diagram as Fig. 1, it is evident that the composition of the vapor in equilibrium with the liquid is required for all pressures, temperatures and concentrations.

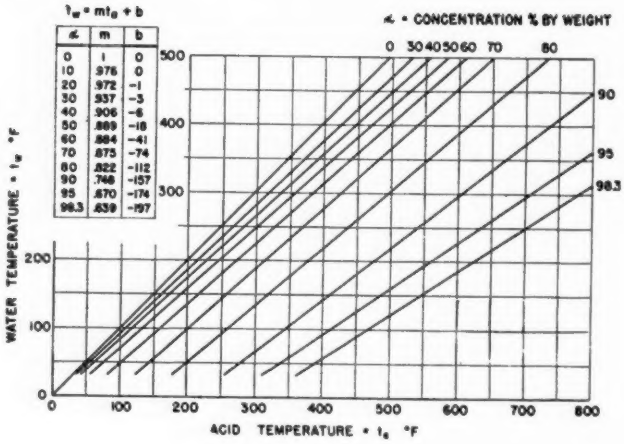
The vapor pressure of sulfuric acid solutions has been tabulated in Rogers (1)† including data for an azeotropic acid-water mixture. This mixture, 98.3 per cent acid, 1.7 per cent water by weight, has the property that the vapor composition is the same as the liquid; hence, it behaves as a pure component.

Thomas and Barker (2) made measurements of the vapor pressures of sulfuric acid-water solutions having high acid concentrations. They determined the composition of the vapor assuming no dissociation in the vapor.

Taylor (3) and Francis (4) made measurements of the dew points of air-water-acid mixtures at atmospheric pressures. Francis made calculations to show the influ-

* By permission of The American Society of Naval Engineers, Inc.

† Numbers in parentheses refer to titles in bibliography at end of article.



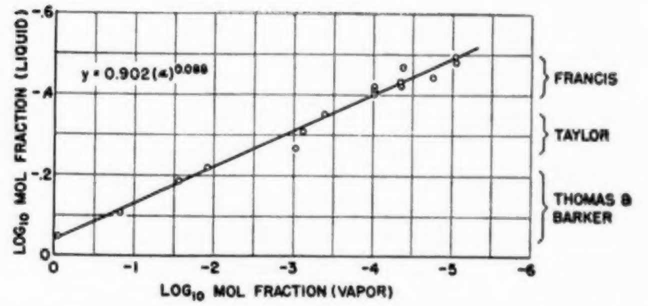


Fig. 3—Sulfuric acid liquid-vapor equilibrium curve for constant pressure, above

Fig. 2—Dühring lines for sulfuric acid, left

ence of gases normally found in combustion products on the water dew point. His data show that the gases may be treated as diluents which fix the partial pressure of the acid-water system and have practically no effect on the dew point.

The variation of the boiling point with pressure for various concentrations of acid-water solutions is conveniently described by the use of Dühring lines (5). The boiling point of a mixture may be determined with respect to that of a known reference liquid. Fig. 2 has been constructed for the sulfuric acid-water system with water as the reference. For example, water boils as 212 F at standard conditions; a 50 per cent by weight acid-water mixture boils at 259 F at the same pressure. In like manner, if the pressure is such that water boils at 150 F, then a 70 per cent by weight acid-water mixture boils at 257 F.

The Dühring lines are assumed to be straight. The results in Table 1 have been calculated from data appearing in Rogers and the International Critical Tables (6).

The data enable construction of the liquid boiling line (as the lower line of Fig. 1) for various pressures and for acid concentrations to 98.3 per cent by weight. Consequently, these data are applicable to combustion gases from usual fuels.

The equilibrium relations between vapor composition and liquid composition are shown in Fig. 3. These data were obtained at atmospheric pressure and less by Thomas and Barker, Taylor, and Francis. With the aid of Fig. 2 and the dew point-vapor composition data of Taylor and Francis, we computed the corresponding equilibrium liquids. Conversion factors for ease in computation are given in Table 2.

It is assumed that the data in Fig. 3 apply to pressures and temperatures other than those at which obtained. Study of the Thomas and Barker data shows that the change in vapor composition with temperature for a given liquid composition is negligible.

Abel (7) used a method of calculating the liquid-vapor equilibrium based on thermodynamic procedures. His results are in agreement with the data of Thomas and Barker, Taylor, Francis, and Fig. 3, the greater variations occurring at vapor concentrations with acid mol fractions of 10^{-5} and less at elevated temperatures.

Taylor also performed experiments with acid vapors in air having differing dew points. The partial pressures of the acid vapors were nearly constant. He measured the strength of condensate and also computed it from vapor pressure data. His calculated and measured results are in excellent agreement, as shown in Table 3.

The composition of the condensing liquid is dependent upon temperature alone, at constant pressure, once condensation has begun. These experiments, therefore, indicate that dynamic systems may behave as static ones.

The experiments also show that the partial pressure of the vapors does not decrease appreciably during condensation even when the difference between the dew point and the condensing temperature is as large as 158 deg F; as for example with his strongest acid vapor—approximately 0.24 per cent acid by weight. It is evident that significant changes in partial pressures may not occur above a temperature as that corresponding to t_3 of Fig. 1.

The Equilibrium Diagram

The sulfuric acid-water equilibrium diagram for acid vapor condensables at a partial pressure of 76 mm Hg, approximating marine boiler operating conditions, is shown in Fig. 4. The figure illustrates how relatively minute amounts of acid in the vapor cause a large dew point elevation and the subsequent high acid strength of the initial condensate. Fig. 4 was constructed from the data of Figs. 2 and 3.

In many cases, construction of the diagram is unnecessary. The partial pressures of the acid vapors may be considered as that of the water in the vapor. From the Thomas and Barker data, for example, it may be seen that with a liquid composition of 89.3 per cent acid, the vapor composition is only 6.2 per cent acid by weight or 1.2 mol per cent. Hence, for relatively high acid concentrations in the liquid, the equilibrium vapor is almost all water, except near the azeotropic mixture.

The strength of the acid deposited may be obtained from the intersection of the water dew point corresponding to the partial pressure and that of the observed dew point in Fig. 2. This may be illustrated with the Thomas and Barker data as follows: their boiling point for 89.3 per cent acid (by weight) is 494 F at standard pressure;

TABLE 3—COMPARISON OF DEW POINT AND CONDENSATE STRENGTH CONDENSING TEMPERATURE 243 F, CALCULATED WATER DEW POINT, 109 F

Dew Point, F	Sulfuric Acid Condensate, Per Cent by Weight Observed, Taylor	Computed Taylor	Computed Fig. 2
320	76.0	75.3	75.6
347	74.9	75.1	75.6
374	75.3	75.5	75.6
401	75.5	75.1	75.6

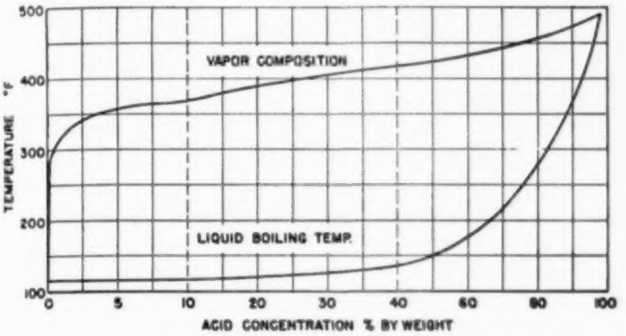


Fig. 4—Equilibrium diagram for sulfuric acid and water. Pressure = 0.1 atm. (76 mm Hg), above

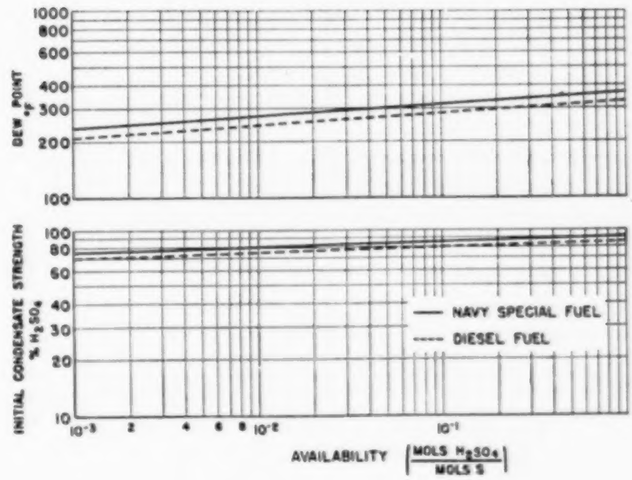


Fig. 5—Influence of the availability of sulfur as sulfuric acid on the dew point and initial condensate strength. Vapor pressure = 0.1 atm, right

that of water is 212 F. Using linear interpolation, the intersection of 212 F and 494 F in Fig. 2 shows the acid strength of the liquid to be 90.1 per cent by weight.

The dew point of a system of gases of known composition and partial pressures may also be predicted without the diagram. The composition of the liquid in equilibrium with a given vapor at the dew point is given in Fig. 3. The intersection of the determined liquid composition with the water dew point in Fig. 2 yields the boiling point of the liquid which is the dew point of the vapors.

It should be noted that the bulk of the gas stream need not be cooled to the dew point to cause condensation. A cold surface in way of the gases may collect condensate, the rate of collection being governed by the laws of diffusion.

If the bulk of the gas is cooled below the dew point, then the partial pressures of the vapors are reduced below that of the initial dew point. However, the amount of condensate may be calculated by assuming that the gas contains water vapors only, and that the partial pressures in the system do not change appreciably until the pure water dew point is reached.

The Availability of Sulfuric Acid

The data presented in the previous sections are applicable when the composition of the gases is known. Francis has made studies to show that sulfuric acid alone could adequately explain the high dew points of flue gas which have been measured. It was mentioned that the usual constituents of flue gas have negligible effect on the dew point as long as the concentration and partial pressure of the acid-water mixture is not changed. The only continuous source of sulfur in the combustion gases is from the fuel. The highest possible concentration and corresponding dew point, therefore, may be computed from the fuel analysis, the humidity of the combustion air, and the fuel-air ratio.

Such a computation assumes complete conversion of the sulfur in the fuel to sulfur trioxide and subsequent mixing with the water vapor in the gas stream. It also assumes that all such converted sulfur remains apart from other combustion products or materials in the gas stream. In the practical case, however, such assumptions may be invalid. A variable, to be spoken of as "availability," is present for each and every combustion application. We choose to define availability as the ratio of mols of sulfuric acid in the gas stream at the place of condensation to the mols of sulfur fired, or

$$\text{Availability} = \frac{\text{mols } H_2SO_4}{\text{mols S}}$$

The availability is dependent upon (1) the oxidation of sulfur and its compounds to sulfur dioxide, (2) conversion of the dioxide to trioxide, (3) adsorption of the dioxide and/or the trioxide on solid surfaces, (4) combination of the oxides with alkali metals in the ash, (5) combination of sulfur trioxide with water vapor in the gas stream, and (6) dissociation of acid vapors.

Manning (8) lists a sequence of reactions involved in the deposition of a sulfuric acid film from a flue gas as follows:

$$SO_2 + {}^1/_2O_2 \rightarrow SO_3\text{, slow except above 1830 F} \quad (1)$$

$$SO_2\ (gas) + H_2O\ (gas) \rightarrow H_2SO_4\ (gas)\text{, fast} \quad (2)$$

$$H_2SO_4\ (gas) + xH_2O\ (gas) \rightarrow H_2SO_4\ (\text{liquid solution})\text{, fast} \quad (3)$$

In the normal combustion process rapid vaporization, high temperatures, and reasonable mixing of the reactants provide conditions for rapid oxidation of sulfur and its compounds in the fuel to SO_2.

The governing condition is reaction (1). An extensive literature exists with respect to the speed of reaction and the equilibrium conversion. In general, the speed of reaction is rapid at high temperatures but equilibrium conversion to the trioxide is small; the speed of reaction is slow at low temperatures but greater conversion occurs.

Manning summarizes reaction (1) as follows: "In the absence of a catalyst, the actual conversion of SO_2 to SO_3 in a flue gas would be small ($^1/_2$–1 per cent) since at the temperature at which reaction was rapid, equilibrium conversion is small. If the gas met a catalyst, the reaction rate would be high at low temperature, and since the equilibrium conversion is high at low temperatures, the actual conversion would be increased." Data in Rogers shows that conversion is complete at 570 F with a gas containing 7 per cent SO_2, 14 per cent O_2, and 79 per cent N_2, by volume, at atmospheric pressure.

The presence of catalysts in the flue gas stream has been studied. Rogers lists the catalysts used in the manufacture of sulfuric acid; predominantly platinum and vanadium types in the contact process and nitrogen

oxides in the chamber process. Manning also states that Fe_2O_3 has little effect in the contact process below 750 F and maximum effect at about 1100 F. Rylands and Jenkinson (9) state: "It has been known for many years that iron sulfate is not only deliquescent, but is also a powerful catalytic agent, that can convert SO_2 to SO_3."

Experiments by Harlow (10) show: (*a*) flue gases passed over heated, rusty mild steel scrap had a higher dew point (attributable to conversion) after exposure than before, greatest conversion occurring at about 1100 F and almost none below 800 F; (*b*) steel coated with pulverized ash was not effective as a catalyst; and (*c*) rust, as Fe_2O_3, is effective as a catalyst while rust, as Fe_3O_4, is not. A major conclusion by Harlow is that the metal temperatures of superheater tubes with their coating of metallic oxides are conducive to conversion.

Rogers also has a discussion of the effect of pressure showing that higher pressures promote conversion.

Corbett, Flint and Littlejohn (11) describe acid dew point measurements made in the gases of coal-fired boilers. Their measurements show that a difference may be expected with a given fuel dependent on the method of firing, pulverized or grate fired; and that the cleanliness of the firesides does not affect the dew point.

The American Gas Association (12) refers to experiments by Johnstone and Maconachie as follows:

Fuel	$R = \frac{SO_3}{SO_2}$, by Volume
Pulverized coal	0.008
Stoker fired coal	0.016–0.029
Petroleum residues and natural gas	0.137
City gas with 95 per cent excess air	0.28

If all the sulfur fired is oxidized and there is sufficient water vapor in the gases to combine with all the SO_3 to form a mixture less concentrated than the azeotropic, then the values, R, given above may be compared with the availability ratio by the equation:

$$\text{Availability} = \frac{R}{1 + R}$$

The difference between the results with pulverized and stoker fired fuels may be due to adsorption of the sulfur oxides by ash particles. If the dioxide is adsorbed, then expected conversion may be diminished with resultant loss of availability. If the trioxide is adsorbed, then loss of availability occurs after conversion. Soot deposits having 6 per cent of the sample weight as free acid have been obtained from oil-fired naval boilers.

The influence of alkali metals in the fuel availability may be judged from an analysis of fireside deposits. Such deposits collected after firing fuel oils contaminated with sea water contain relatively large amounts of sodium sulfate. The high deposition temperature of such salts could reduce the number of sulfate ions available for possible deposition at the acid dew point.

Reaction (2) is listed as fast, and according to Taylor, the equilibrium conditions permit the reaction to be complete to the right below 428 F. Reaction (3) is also fast. Experiments of Taylor show a rapid equilibrium.

The observations and measurements presented have the effect of lowering the dew point as calculated above. The amount of reduction is dependent upon the availability. With a known availability ratio, and the calculated maximum acid content of the flue gas for a particular combustion process, the dew point of the gases along with the strength of the initial condensate may be determined from Fig. 4.

The change of dew point and initial acid condensate strength with availability is shown in Fig. 5. The data for construction of the figures are:

a. Navy special fuel: 86.5 per cent C, 11.1 per cent H, 2.4 per cent S by weight.

b. Navy specification diesel fuel: 86.0 per cent C, 13.6 per cent H, 0.4 per cent S by weight.

c. Atmospheric humidity arbitrarily chosen to fix the partial pressure of condensable at 0.1 atm.

At unit availability, for the example chosen, the strength of acid in the condensable is 5.8 per cent by weight for the Navy special fuel and 0.9 per cent by weight for the diesel fuel. In this case, the initial condensate strengths, from Fig. 5, are 91 per cent and 85 per cent, by weight, respectively. If the gases are sufficiently cooled below the dew point, therefore, the condensate strength will vary between 91 per cent and 5.8 per cent for the Navy special fuel, and between 85 per cent and 0.9 per cent for the diesel fuel.

It may be seen that the reduction in dew point, compared to an availability of unity, is less than 100 deg F when the availability is 10^{-2}. In new designs it may be necessary to assume this reduction to be small, and to base calculations on maximum availability, recognizing that the error involved is on the side of safety.

A convenient method to determine the availability in a given installation combines the use of a dew point meter and a knowledge of the highest possible concentration of the acid vapors in the gas stream. The amount of acid in the vapors is relatively small and the computed water content may be used to fix the partial pressure of the condensables.

BIBLIOGRAPHY

(1) Furnas, C. C., editor, "Rogers Industrial Chemistry," Sixth Edition, Vol. One.

(2) Thomas, J. S., and Barker, W. F., "The Partial Pressures of Water Vapor and of Sulfuric Acid Vapor over Concentrated Solutions of Sulfuric Acid at High Temperatures," *Journal of the Chemical Society*, Vol. 127, 1925, p. 2820.

(3) Taylor, H. D., "The Condensation of Sulfuric Acid on Cooled Surfaces Exposed to Hot Gases Containing Sulfur Trioxide," *Trans.* of the Faraday Society, No. 346, Vol. 47, Oct. 1951, p. 1114.

(4) Francis, W. E., "The Measurement of the Dew Point and H_2SO_4 Vapor Content of Combustion Products," The Gas Research Board Communication GRB64, June 1952.

(5) Walker, W. H., Lewis, W. K., McAdams, W. H., and Gilliland, E. R., "Principles of Chemical Engineering," Third Edition, 1937, p. 412.

(6) "International Critical Tables," 1st Edition, 1928, p. 203.

(7) Abel, E., "The Vapor Phase above the System Sulfuric Acid-Water," *Journal of Physical Chemistry*, Vol. 50, 1946, p. 260.

(8) Manning, G. H., Comment on Reference (10), *Proceedings* of the Institution of Mechanical Engineers, Vol. 151, 1944, p. 306.

(9) Rylands, J. R., and Jenkinson, J. R., "Bonded Deposits on Economizer Heating Surfaces," *Proceedings* of the Institution of Mechanical Engineers, Vol. 151, 1944, p. 291.

(10) Harlow, W. F., "Causes of High Dew Point Temperatures in Boiler Flue Gases," *Proceedings* of the Institution of Mechanical Engineers, Vol. 151, 1944, p. 293.

(11) Corbett, J., Flint, D., and Littlejohn, R. F., "Developments in the B.C.U.R.A. Dew Point Meter for the Measurement of the Rate of Acid Build-up on Cooled Surfaces Exposed to Flue Gases," *Journal of the Institute of Fuel*, Vol. 25, No. 146, Nov. 1952, p. 246.

(12) "Fuel-Flue Gases," American Gas Association, 1940, New York, p. 113.

The Effect of Measurement Errors on Plant Performance Tests*

Accepting and approving newly installed equipment requires full understanding of the various test limitations and instrument accuracies since the initial performance tests prove out not only design engineers' predictions, but manufacturers' guarantees

By DR. S. BARON

Head, Heat Balance Dept., Burns and Roe, Inc.

WITH the cooperation of both clients and equipment manufacturers, Burns and Roe, Inc. has been conducting performance tests on newly installed plants for a number of years. The tests cover principally major equipment performance under actual operating conditions. The results of the tests are vitally important to all concerned since they permit: (1) checking operating station thermal efficiency with that predicted; (2) determining the performance and operation of individual equipment and comparing the results with those guaranteed by the manufacturers; (3) checking design calculations of pressure drops in the major systems with the actual performance; (4) determining station auxiliary loads, and (5) appraising any new developments in design.

Station instruments serve wherever feasible. But preceding the tests, all bourdon type pressure gages are checked out with deadweight testers. We use mercury in glass thermometers for temperature readings below 400 F and check them out first in boiling liquids and/or ice. For temperatures above 400 F, iron constantan thermocouples are used after a check-out similar to the mercury thermometers. A calibrated feedwater nozzle is a permanent part of the usual plant installation and we use it to determine the feedwater flow. Either a high-pressure manometer is installed across the feedwater nozzle or we employ the operating flow recorder provided it is checked out by water column or weights. If coal weights are to be taken, we also check the coal scales. The station watt-hour meter is calibrated with a standard before running the tests. The above procedures, while routine, are necessary.

Instrumentation Errors

The instrumentation errors in this paper represent the limit of possible errors anticipated with the type of instruments discussed provided, of course, that these instruments are in good operating condition. Further, the magnitude of the errors as stated herein are generally those of the ASME Power Test Codes. In cases where magnitudes of errors in some readings are not explicit in the Power Test Codes, we have assigned reasonable values based on experience. The following paragraphs enumerate the major errors of the various instruments involved with the performance tests of the major plant equipment.

Flow Readings

Various primary elements are possible for measuring flows such as weighing tanks, volumetric tanks, nozzles, orifices and gravity tanks. Of this group nozzles or orifices prove the most convenient. According to the ASME Turbine Power Test Code, the limits of possible error of these primary elements are:

Weighing tanks = ±0.25%
Volumetric tanks = ±0.50%
Steam nozzles and orifices = ±1.50%
Feedwater nozzles and orifices = ±1.25%
Gravity tanks = 0.50%

Since a feedwater nozzle installation and calibration run employs weighing or volumetric tanks, the possible error in the nozzle would be comparable to that of the tanks. With a number of test points taken for calibration purposes and a mean curve drawn through the test points, the error in a calibrated nozzle could be expected to be 0.25% or less. Calibrating the transmitter and recorder with water column, deadweight tester, or dummy load admits a possible error of ±0.25% of full scale. A manometer across the orifice or nozzle would give rise to an error of 0.5% in pressure differential or 0.25% in flow. Furthermore, additional reading error of 0.15% of full scale could be expected with strip charts and 0.4% of full scale with round charts. Should readings fluctuate, the visual averaging would introduce somewhat higher errors. The probable error (based on the square root of the sum of the squares of the errors) for a calibrated nozzle or orifice would be ±0.35% with manometer and ±0.55% with recorder. For purposes of discussion and evaluation, an average error in flow reading of ±0.50% would be reasonable at maximum flows.

Pressure Readings

According to ASME Power Test Code, for pressures above 35 psia, calibrated bourdon or deadweight gages should be used. The maximum instrument error for these gages is 0.5% of full scale or 1.0% at half scale which is the usual operating reading. For pressures below 35 psia, ASME recommendations call for mercury manometers which can be read to 0.01-in. Hg. Gener-

* Presented under the same title before the New York Section, Regional Conference, Instrument Society of America, Feb. 4, 1954, New York, N. Y.

ally, due to fluctuations in readings, the best that could be expected would be 0.05-in. Hg. Exhaust pressure measurements by absolute gages or by combination barometer and vacuum readings would have probable error of 0.05-in. Hg due both to fluctuations in reading and to variations in pressure throughout the exhaust section.

Temperature Readings

According to the ASME Power Test Code, temperature readings should be accurate within 0.5% above ice point. Mercury-in-glass thermometers of test grade type fall within this range. Thermocouples of iron constantan type have a possible error of $\pm 0.5\%$ in emf and $\pm 0.2\%$ in potentiometer or a probable mean of $\pm 0.55\%$. Circulating water temperature measurements by glass thermometers can be read to 0.1 F. These thermometers are checked against each other at a number of temperature points before the tests, and correction factors applied to them.

Coal and Combustion Readings

The ASME Power Test Code on Steam Generators requires that coal scales be checked prior to and after the test to an accuracy of 0.2%. In the case of the heating value determination of coal, there would be a probable error of 0.3% for the same observer and 0.5% for different observers. In the Orsat analyses of exit gases, a possible error of 0.2 cc exists in the burette bore and 0.2 cc error arises in reading carbon dioxide and oxygen. These errors combine to give an error of 3% in the calculated weight of dry gas per pound of fired fuel.

The ultimate analysis of coal demands very precise measurements and requires a highly trained technician to obtain reliable results. The method consists of oxidizing the sample in an electric furnace and absorbing the produced carbon dioxide and water with potassium hydroxide and calcium chloride, respectively. The change in weights of absorbents is a measure of the carbon and hydrogen content of the coal. An error of 1% in analyses of carbon and hydrogen is probable even with a first-rate technician. A 1% error in carbon analysis would account for 1% error in dry gas loss calculation and 1% error in hydrogen analysis would account for 9% error in calculation of heat loss of water formed in combustion. Furthermore, a 1% error in coal moisture would give an error of 1% in the calculation of heat loss of water in the coal.

How these instrumentation errors affect the calculated test results shows under the headings below for the equipment tested.

Electrical Readings

The current and potential transformers used with the station watt-hour meter to obtain generator output should be calibrated. The calibration curves should be available with the watt-hour readings. Uncalibrated transformers have been known to be 1% off. Preceding the test, the station watt-hour meter should be calibrated "in place" with a standard. The error under these conditions would be about 0.1%.

Where portable meters measure power input to motors, the following errors would be anticipated in accordance with ASME Power Test Codes:

Wattmeter with auxiliary apparatus = 0.25–1.0%
Watt-hourmeter with auxiliary apparatus = 1–2%
Voltmeter with auxiliary apparatus = 0.25%
Ammeter with auxiliary apparatus = 0.5–1.0%
Power factor with auxiliary apparatus = 1.0%

Speed Readings

Frequency errors in turbine generators run less than 0.1 cycle in a 60-cycle machine. Where pump and motor speeds are measured with a hand counter, the expected error would be 1.0%. The d-c type speed measuring devices have errors of 20 rpm and additional error of 0.5% for every 20 deg F change in temperature. The a-c type measuring devices have probable errors of 50–100 rpm.

Steam Turbines

The turbine heat rate is the measure of the performance efficiency of modern extraction turbines. The guaranteed figures, as specified by the turbine manufacturers, or calculations based on specified turbine performance, depend on a number of conditions involving steam and performance of auxiliary equipment. Should any of the conditions, during tests, be any different than specified conditions, the turbine heat rate would then vary. Some of the factors that must be "tied down" to make test values comparable to guaranteed values are throttle temperature and pressure, reheat temperature, reheat pressure and pressure drop, exhaust pressure, number of heaters, heater performance, final feedwater temperature, bleed pressure and pressure drops, generator load and power factor, and miscellaneous leakages of steam and water in the cycle.

The turbine heat rate in a reheat cycle is generally defined as the total heat picked up by the feedwater evaporated to steam in the boiler, plus the heat picked up in the reheater, divided by the gross generation measured at the generator terminals. According to ASME those cycles which consider the heat recovery from the boiler feed pump, add this heat to the numerator of the turbine heat rate expression. Turbine manufacturers, on the other hand, subtract the pump power from the gross generation. It is, of course, very important that the test turbine heat rates be corrected to the same basis as those on which the guaranteed turbine heat rates were predicted.

No attempt is made to regulate back pressure or throttle and reheat conditions beyond their normal operation. However, test loads are chosen corresponding to the "valve points" on the turbine.

Errors in instrumentation give rise to two types of errors in calculating test turbine heat rates. One of these errors, that appearing in measuring flows, generation, and enthalpy, can produce an error in the calculated turbine heat rate. Whereas, the other type of instrument error, such as occurs in evaluating throttle temperature and pressure, reheat temperature, back pressure, reheat pressure drop, bleed steam pressure drop, heater performance, condenser performance, power factor, and

leakages produces cycle variations and a combination of errors. For example, if the true throttle temperature is 1000 F and the instrument reading shows 1005 F, the calculated turbine heat rate not only will be higher than the true value, but, also, a correction should be made for the fact that the higher throttle temperature would give a better turbine heat rate than possible at 1000 F. Correction curves are generally supplied by the turbine manufacturer for deviations in throttle temperature, throttle pressure, reheat temperature and back pressure. Corrections for deviation between test and actual, for pressure drops, leakages, and heat exchanger performance, can be estimated by heat balance calculations. Table I lists the expected errors in the calculation of the test turbine heat rate, arising from errors in the test instruments. Specifically, the unit considered in Table I is a 62,500-kw turbine generator with throttle conditions of 1650 psig, 1000 F, reheat temperature 1000 F, back pressure 1 in. Hga, and a six-heater cycle.

Considering all the errors enumerated in Table I the probable error in calculating the turbine heat rate would be about 1%. Test results reported by turbine manufacturers based on carefully conducted tests, have shown results to be on the average of 1% better than guaranteed. The American turbine manufacturers include a certain percentage tolerance in their guarantees to protect themselves from errors in design, shop tolerances and test instrumentation.

It may appear that this 1% error is small and unimportant. However, this 1% evaluated as an annual fuel bill, represents a very costly item. As an example; should the guaranteed turbine heat rate be off 1%, which is about 100 Btu/kwhr on overall station heat rate, and with fuel cost at $0.30/million Btu's, a net generation of 60,000 kw for 8000 hours' operation in a year would cost, fuelwise, $14,400. Capitalized at 10%, this fuel cost would justify a $144,000 investment. A carefully conducted turbine test would be far cheaper and should it uncover a 1% poorer efficiency, would easily justify its cost.

Table I is based on tests at optimum load. As test loads decrease, it is probable that the percentage error in many of the instruments would increase. Should the errors be inversely proportional to the test load, the additional fuel cost for poor efficiency at low loads would be comparable to that of the maximum loads.

TABLE I—TURBINE TEST

Measurement	Instrument error	Error in Calculated Turbine heat rate	Error due to Cycle Deviation	Total Possible error
Throttle temp.	±0.5%	±0.3%	±0.1%	±0.4%
Throttle press.	+1.0%	±0.0%	±0.2%	±0.2%
Reheater outlet temp.	±0.5%	±0.2%	±0.1%	±0.3%
Reheater inlet temp.	±0.5%	±0.2%	...	±0.2%
Final feedwater temp.	±0.5%	±0.2%	...	±0.2%
Back press.	±0.05-in. Hg	...	±0.1%	±0.1%
Flow	±0.5%	±0.5%	...	±0.5%
Generation	±0.1%	±0.1%	...	±0.1%
Bleed temp.	±0.5%	±0.0%	...	±0.0%
Bleed press.	±1.0%	±0.0%	...	±0.0%
Feedwater and drain temp.	±0.5%	±0.0%	...	±0.0%
Reheater press. drop	±2.0%	...	±0.2%	±0.2%
Bleed steam press. drop (all heaters)	±2.0%	...	±0.1%	±0.1%
Power factor	±0.05	...	±0.1%	±0.1%
Heater terminal difference (all heaters)	±1.0%	...	±0.2%	±0.2%
Condenser subcooling	±0.05-in. Hg ±0.1 F	...	±0.0%	±0.0%

Steam Generators

The two methods used for determining boiler efficiencies are: (1) heat input–heat output method and (2) heat loss method.

In method 1, the heat input can be determined from the weight of fuel fired and its heating value. The heat output then is determined from the quantity of steam generated, the enthalpy rise from the water stream entering the boiler to steam flow of the superheater outlet and reheat flow (if any) and its enthalpy rise through the reheater. If any other stream, such as spray attemperation, enters the boiler, its heat should be included in the output figure. The ratio of heat output to heat input, times 100, would be the boiler efficiency.

In the second method, all the heat lost from the boiler per pound of fired fuel, is determined and boiler inefficiency then calculated. The difference between 100% and the inefficiency will, of course, be the boiler efficiency. This method requires sampling and analysis of the fuel, flue gas, flue dust and refuse; measurement of flue gas temperature and inlet air temperature, moisture in the air and loss due to radiation and convection.

The heat input–heat output method enjoys ASME Power Test Code preference while the boiler manufacturers favor the heat loss method. In general, the guarantees check out more easily by the heat loss method than by the heat input–heat output method since the losses measured are those set forth in the boiler manufacturer's proposal.

In both methods of determining boiler efficiency, sampling errors prove the most critical factor, outside of measurement errors. In determining the fuel heating value, for example, a representative sample of the fired fuel must be obtained. Even after a proper sample has been taken, it is necessary, particularly with coal, to prevent moisture pickup or loss. Proper sampling of flue gas is also difficult due to stratification. The same difficulty arises in obtaining proper sampling of the flue gas temperatures. Proper sampling of the refuse and flue dust represents still another difficulty. Since refuse has coarse and fine particles which, in all probability, have various heating values and since obtaining representative dust samples is difficult, losses from these factors are assumed by setting a value for unburned combustibles in refuse and flue dust based on past experience.

In the heat loss method, a number of losses need to be assumed or neglected. The loss due to radiation and convection, for instance, comes off a curve developed by the American Boiler Manufacturers Association. The loss due to combustibles in the ash and slag generally draws an assumed value based on the boiler manufacturer's experience. The loss due to sensible heat of the refuse is neglected. Similarly, in the case of liquid or gaseous fuels, the heat losses due to unburned hydrocarbons and hydrogen are generally neglected.

Table II lists the instrument and analysis errors involved with the two methods and indicates to what extent the calculation of boiler efficiency is affected by these

errors. Although the probable error of the heat loss method is ±0.3% and that of heat input–heat output method is ±0.8%, there are additional errors in sampling and unaccounted for losses in the heat loss method that in actuality, make the heat loss method comparable if not the poorer of the two. If the procedure described in the ASME Power Test Code for testing boilers by the heat loss method is rigorously followed, particularly with respect to sampling of flue gas, the results would probably be as reliable as the heat input–heat output method.

The boiler manufacturers in stipulating the guaranteed boiler efficiency include, in their tabulation of inefficiencies due to heat losses, a value of 1.5% for unaccounted for losses. This percentage then represents the manufacturers' tolerance in their guarantees to protect themselves from errors in test and design. In a test by the heat loss method, the 1.5% unaccounted for loss does not appear as it is assumed that all losses have been considered.

In evaluating the effect of boiler efficiency on the annual fuel bill, a 1% change in efficiency would be comparable to a 1% change in turbine heat rate and therefore, would be worth $14,400 annually for the sample case cited under steam turbines.

TABLE II—BOILER TEST

Measurement	Instrument or Analysis Error	Error in Calculated Boiler Efficiency
Heat Input–Heat Output Method		
Superheater outlet temp.	±0.5%	±0.3%
Superheater outlet press.	±1.0%	±0.0%
Reheater outlet temp.	±0.5%	±0.2%
Reheater inlet temp.	±0.5%	±0.2%
Final feedwater temp.	±0.5%	±0.2%
Feedwater flow	±0.5%	±0.5%
Calculated reheat flow	±0.5% temp. ±1.0% press.	±0.0%
Coal scales	±0.2%	±0.2%
Coal heating value	±0.3%	±0.3%
Heat Loss Method		
Heating value	±0.3%	±0.0%
Orsat analysis	±3.0%	±0.3%
Exit gas temp.	±0.5%	±0.0%
Inlet air temp.	±0.5%	±0.0%
Ultimate analysis of coal		
Carbon	±1.0%	±0.1%
Hydrogen	±1.0%	±0.1%
Moisture in coal	±1.0%	±0.0%

Condensers

The performance of a condenser is determined by comparing actual shell side pressure with the expected pressure based on the test heat duty and known design of the condenser. The expected pressure can be calculated from the basic heat transfer equation:

$$Q = UA\Delta$$

where, Q, the condenser heat duty is determined from the test; A, the heat transfer area of the condenser is known from the design; U, the heat transfer coefficient is taken from the Condenser Standard of Heat Exchange Institute. U is dependent upon condenser inlet circulating water temperature, circulating water velocity, tube size and cleanliness factor. The water velocity is determined from the condenser heat duty and the rise in temperature of circulating water. Condenser inlet circulating water temperature is taken during the test, the tube size is known from the condenser design and the cleanliness factor is generally assumed at 0.85. Δ, the logarithmic mean temperature terminal difference, is calculated from the known Q, A, U. Since

$$\Delta = \frac{t_1 - t_2}{2.3 \log \dfrac{t_v - t_1}{t_v - t_2}}$$

t_1 is inlet water temperature, t_2 is outlet water temperature and t_v is steam temperature in shell. Values of t_1 and t_2 are determined from test, Δ by calculation, and therefore t_v can be determined from the above equation. The shell pressure then corresponds to t_v, the saturated steam temperature.

The value of Q can be determined by test by either of two methods. In one method a heat balance is carried out whereby the heat duty of the condenser is determined by the difference between the heat output of the boiler and the energy output of the turbine.

In the second method, the enthalpy of the steam entering the condenser is determined by measuring the temperatures and pressures of the steam bleeds of the turbine, plotting these points on a Mollier Chart and extrapolating the points to the shell pressure. The actual enthalpy of the steam is somewhat higher due to leaving losses of the steam and can be determined from the exhaust loss curve supplied by turbine manufacturer. Knowing, then, the steam enthalpy, and measuring the condensate flow and temperature leaving the condenser, it becomes possible to calculate the condenser heat duty. Both methods have about the same degree of accuracy as flows, pressures and temperatures must be measured in either case.

The error in heat duty of the condenser can be calculated from the errors for the turbine heat rate enumerated in Table I. Since the heat input to the turbine is about one third of the heat generated, the error in generation in so far as it affects condenser heat duty would be reduced by one third.

Two other critical readings which have an important bearing on the condenser test are circulating water temperatures and condenser shell pressure. The inlet circulating water temperature can be sampled easily enough at some point in the inlet water box or at the discharge of the circulating water pumps. However, careful readings should be taken of the circulating water temperature leaving the condenser. Due to variation in flow through the individual condenser tubes and different heat transfer rates in the various tubes, the temperature of the circulating water leaving each tube would vary somewhat. Therefore, unless there is perfect mixing of the effluent from the condenser, the measured water temperatures would be incorrect. Furthermore, since the rise of circulating water temperature is from 10 to 15 deg F at high loads (and lower at low loads), the thermometers should be accurate and read within 0.1 F.

The pressure readings of the condenser by absolute pressure gage or vacuum gage are very sensitive to leaks and to velocity head of the steam. Taps in the side of shell should be perpendicular to the wall and taps extending into the turbine exhaust neck or the condenser inlet should have guide plates.

Table III demonstrates to what extent the calculated condenser shell pressure is affected by measurement er-

rors. The condenser considered in this table is based on design duty of 250×10^6 Btu/hr, 60 F inlet water temperature, 10 deg F rise in water temperature, and a back pressure of 1.0-in. Hga.

As the results of Table III show, the errors in heat duty and temperature rise of circulating water have a negligible effect on the error in the calculated shell pressure. This can be understood by observing that an error in Q would give the same error in the calculated water velocity. Since U is proportional to the square root of velocity, the error in U would be proportional to the square root of the error in velocity or Q. Since $\Delta = Q/UA$, the error in calculating Δ is only proportional to the square root of the error in Q.

The use of a cleanliness factor raises greater uncertainty in the analysis. The condensers are generally designed for 85% cleanliness which means that the condenser surface is 15% greater than necessary for a 100% clean condenser. With present methods of chlorinating circulating water, screening of the water and high velocities the condensers are generally over 85% clean. The manufacturers' tolerance is then that percentage between 85% and the actual operating cleanliness which can be as much as 100%. Should the condenser perform with 100% cleanliness instead of 85%, the shell pressure would be 0.94-in. Hga instead of 1.00-in. Hga. The cleanliness factor would then allow the tolerance necessary for instrumentation errors. There is, of course, further tolerance in the values of U used in condenser design.

TABLE III—CONDENSER TEST

Measurement	Instrument Error	Error in Calculated Expected Shell Pressure
Heat duty	±1.0%	±0.003-in. Hg
Temp. rise of circulated water	±0.2 F or ±2.0%	±0.003-in. Hg
Condenser shell pressure	±0.05-in. Hg	±0.050-in. Hg

Pumps and Fans

The major items tested under this category are boiler feed pumps, condensate pumps, circulating water pumps, induced draft fans, forced draft fans and ventilating fans. In testing the performance of these pumps and fans the usual practice involves comparing the operating heads and efficiencies with the expected values for various flow rates. The efficiency of pump or fan can be calculated from the ratio of water, air or gas horsepower to the shaft horsepower. The water, air or gas horsepower, in turn, is determined from the measured fluid flow rate and total head developed. The shaft horsepower results from calculations involving the measured power input to the motor after taking into account the motor and coupling (if any) efficiencies. If turbine drives are used, the steam flow to the turbine and the inlet and outlet enthalpy of steam must be measured.

In order to determine the performance of the pumps and fans under discussion, it is necessary to measure the following items:

1. Flow rate of fluid leaving pumps or fans
2. Discharge head of the pumps or fans
3. Suction head of the pumps or fans
4. Speed of pumps or fans and motors
5. Power input to pumps or fans, and
6. Temperature of fluid entering pumps or fans.

In testing boiler feed pumps, the feedwater flow as measured with a calibrated nozzle will have a probable error of 0.5%. The circulating water flow measurement based on heat balance calculations and circulating water temperature rise would have a probable error of 2.1%. Condensate flow based on calibrated nozzle or orifice has a probable error of 0.5% and, with an uncalibrated nozzle or orifice, about 1.3%. Gas flows are generally determined by pitot tube measurements since nozzles or orifices produce high head losses. However, if pitot tube readings are to be reliable, it is necessary that proper sampling be used. The standards of the National Association of Fan Manufacturers have fully covered the test procedure requirements. The probable error in gas flow measurements would be about 0.5%.

In measuring the discharge and suction heads of boiler feed pumps and discharge heads of condensate pumps bourdon type gages with about 1% error are quite satisfactory. For discharge heads of circulating water pumps and suction heads of condensate pumps, mercury manometers should be used. The reading error would be about 0.1-in. Hg due to head fluctuations. For a 10-in. Hg head reading, the error would be about 1%. Circulating water pumps of the axial, propeller, or mixed flow type require suction water level measurements by means of a scale. Centrifugal type circulating water pumps require mercury manometers for pump suction. The error in suction head readings of circulating water pumps would be 1.0%. Head readings for gases are usually determined by water or kerosene filled manometers. These manometers can be read to about 0.1-in. which would be 1% error for a 10-in. reading.

It should be remembered that manometers and pressure gages give static head readings. To determine the test total head, it is necessary to add the velocity head to the static head. In the case of high head pumps such as boiler feed pumps and condensate pumps, the velocity head is negligible. For low head pumps such as circulating water pumps and for fans, the velocity head becomes important.

Since we are primarily interested in the total head, which is the difference in discharge and suction head, the error in the larger of the two head readings would have the greater bearing on the total head. In the case of boiler feed pump and condensate pumps, the discharge head readings rank as the critical values. With circulating water pumps, the suction and discharge readings have about equal importance; therefore, errors in these readings would affect the total head. The discharge head of forced draft fans and ventilating fans are the critical readings while in the case of the induced draft fans the suction head is the more important. In general, the value of total head would have an error of about 1% in pumps and fans. The total head error in circulating water pumps would be about 1.5%.

Fan and pump speeds are related to the heads, capacities, and horsepowers by the "Fan Laws" where: (1) capacity is proportional to speed, (2) head is proportional to square of speed and (3) horsepower is proportional to cube of speed.

Therefore, it is important to remember that in com-

paring test results with expected results, do so at the same speed. A change of 1% in speed would reflect in a 1% change in capacity, a 2% change in head and a 3% change in horsepower, all at constant efficiency. With variable speed operation, the test head and capacity must be corrected to design speed basis for comparison purposes. Pump and fan efficiencies even for variable speed operation are generally based on design speed. Therefore, the efficiencies at test conditions must be compared to the expected efficiencies corresponding to the flows calculated from the test flows after correction by the ratio of design speed to test speed.

In variable speed operation, the pump and fan speeds are necessary to determine coupling efficiencies. The coupling efficiency (excluding small fixed losses) is directly proportional to slip. Therefore, in calculating shaft horsepower of a pump or fan from the power input to the motor, the coupling efficiency must be determined from the motor and pump or fan speeds. A 1% error in speed measurement would affect the shaft horsepower by 1%.

The power input to a motor is generally determined by a portable wattmeter or watt-hour meter. Although the efficiencies of current and potential transformers are close to 100%, there have been known cases in which their efficiencies were off by 1–2%. It is therefore advisable to install calibrated transformers as well as calibrated motors. Power input to the motor can also be determined by taking the voltage, amperage and power factor. This method, however, has a greater probable error than the direct power measurement.

The expected heads of pumps are generally expressed as feet of fluid transferred while the heads of fans are in inches of water. In order to convert the pressure in psi and inches of mercury to feet of fluid, it is necessary to determine the fluid density from the measured fluid temperature. The temperature must also be measured for correcting flows which have been determined by differentials across nozzles or orifices. Errors in temperature measurement will have a negligible effect on the results.

Table IV contains the instrumentation errors involved with pump and fan tests and indicates to what degree the calculations of efficiencies are affected. Considering the case in which the flow rate error is ±0.5%, the probable error in efficiency for variable speed operation would be 2.1% and for constant speed 1.5%. For the case with a 2.1% flow rate error, the probable error at variable speed is 2.9% and 2.5% at constant speed.

TABLE IV—PUMP AND FAN TESTS

Measurement	Instrument Error	Error in Calculation of Efficiency: Constant Speed	Error in Calculation of Efficiency: Variable Speed
Flow rate	±0.5%	±0.5%	±0.5%
	±1.3%	±1.3%	±1.3%
	±2.1%	±2.1%	±2.1%
Total head	±1.0%	±1.0%	±1.0%
Speed of pump or fan	±1.0%	...	±1.0%
Speed of motor	±1.0%	...	±1.0%
Power input to motor	±1.0%	±1.0%	±1.0%

Heat Exchangers

The types of heat exchangers that are tested are tube and shell feedwater heaters, contact heaters, evaporators and evaporator-condensers. These heat exchangers are part of the feedwater or condenser heating cycle and therefore the thermal efficiency of power generation is dependent on the temperature terminal conditions that exist.

The procedure for testing these units is simple enough. For the feedwater heaters, evaporator-condensers and contact heaters, the terminal difference between the water effluent from the heater and the saturated temperature corresponding to the shell pressure is compared with the expected terminal difference for the same water flow. Feedwater heaters with integral drain coolers are tested by comparing the temperature terminal difference of drains leaving the heater and water entering with anticipated terminal difference. For evaporators, the temperature terminal difference of the saturated temperature corresponding to steam pressure and the temperature of the boiling liquid in the shell are compared with the expected for the same rate of makeup to the evaporator. All the above procedures are straightforward and require no special precaution.

Errors in flow measurements are not too critical. The important readings are steam pressures and water temperatures. As discussed earlier, temperature errors can be in the order of ±0.5% and errors in pressure of ±1.0%. The ±0.5% iron constantan thermocouple error is equal to ±2.5 deg F error at 600 F water and ±1.0 deg F error at 200 F water. The ±1.0% pressure error for 600 F level is equivalent to ±1.3 deg F and for 200 F level is equivalent to ±0.4 deg F. The probable error then in temperature terminal difference at 600 F level would be ±2.8 deg F and ±1.1 deg F for 200 F level. The shell side of heat exchangers should be properly vented otherwise shell pressures would include the partial pressure of air which then affects the heat transfer and gives inaccurate temperature terminal differences.

Discussion

It should be emphasized that tests conducted under operating conditions will have greater probable errors than those conducted by equipment manufacturers on their own equipment in their test laboratories. This would be expected since the laboratory tests are conducted under controlled conditions with instruments of greater accuracy than those generally used in tests. However, it is good practice to conduct operating tests of the installed equipment in order to verify that it had not been damaged internally in any time between the laboratory tests and operation. Furthermore, periodic testing of the equipment permits a running check on the operating performance and allows the operators to determine whether any particular item needs to be overhauled or repaired during a scheduled shutdown. In this way, an unscheduled shutdown which is very costly, would be avoided.

Over and above the immediate advantages just mentioned are the indirect ones gained from developing operator familiarity with equipment and performance.

Reactions of Salts in Boilers

By R. S. YOUNG*

Citing investigations by Ipatieff, the author suggests that certain types of internal deposits on boiler metal, including copper, may be explained on the hypothesis that hydrogen under pressure has the property of displacing metals and metal oxides from solution. The idea is not altogether new but is susceptible to profitable discussion.

When power station boilers are inspected deposits of several types may occasionally be found at points within the drums and tubes. These deposits may be crystalline or amorphous, and occur in a variety of forms and colors. A study of certain characteristic yellow, green, or brown nodules which appeared when conditioned water was allowed to cool and stand some time in a boiler after taking the latter off range, has shown that phosphate and sulfate together are necessary for the formation of such barnacles (4). Laboratory tests on the effect of conditioning reagents in boiler water on steel specimens gave the identical type of nodule with the accompanying underlying pitting found in actual practice when conditioned water is allowed to cool and remain undisturbed for a considerable time in a boiler after a long steaming period. These deposits were assumed to arise by the formation of sparingly soluble compounds from chemical interactions between conditioning agents and iron.

In laboratory tests, and probably in many cases of boiler operation, this is undoubtedly the mode of formation of most nodular deposits. It is possible, however, to advance another explanation for the origin of certain boiler deposits, namely, that they might be due to the displacement of metals or metal oxides by hydrogen. This is offered solely as an hypothesis without supporting experimental data. The latter would be extremely difficult to obtain, in view of the high pressures and temperatures in the modern boiler.

V. N. Ipatieff, the great petroleum chemist, whose contributions to catalytic cracking and the production of high octane gasoline have meant so much to the United States and to the world, while still in his native Russia, carried out over many years some very interesting work on the effect of hydrogen under pressure and temperature on inorganic reactions (3). He found that for each metal there is a certain critical temperature and pressure which permits the displacement of the metal from its solution by means of hydrogen. Contrary to theoretical calculations based on thermodynamic considerations, Ipatieff found that these pressures were moderate. For instance, the calculated required pressure of hydrogen for the separation of cadmium from its nitrate is 10,000 atmospheres, yet Ipatieff found that a 2 N solution of this salt at 270 C and 220 atmospheres pressure yields hexagonal crystals of cadmium in 10 hours.

At 90 C and 25 atmospheres pressure, copper sulfate gave initially crystalline $CuSO_4 \cdot 2Cu(OH)_2$. On longer contact the separation of Cu_2O occurred, while after two days crystals of copper separated out along with Cu_2O. After seven days heating the only product was metallic copper.

Metallic copper has been found in boiler systems, and was assumed to have originated from copper, brass, or bronze, fittings in contact with boiler water condensate or evaporator makeup. It has always been difficult to visualize, however, how it could be deposited from the alkaline boiler water. It is only in acid solution that copper will precipitate on the surface of iron or a metal higher in the electromotive series. With hydrogen under a pressure of 375 psig, which is about that found in a medium pressure boiler, we have an easy explanation for the deposition of copper under these conditions.

Ipatieff found the precipitation of crystalline basic salts and metal oxides by hydrogen pressure took place by means of hydrolysis, and in this case water played an oxidizing role which increased with rise of temperature. The critical temperature of precipitation of metallic iron is above 360 C and under these conditions water tends to oxidize the iron back to an oxide. A solution of iron nitrate, for instance, produced octahedral crystals of Fe_3O_4. It is probable that other iron solutions under hydrogen pressure will likewise give the minute black crystals of magnetite sometimes seen on the surface of boiler drums and tubes. Ferric phosphate was found to produce crystalline basic salts of iron. The products had the general formula $Fe_n^{+++}Fe_m^{++}(PO_4)_x.yH_2O$, and depending on the composition had various colors ranging from green to blue to black.

His experiments were carried out in a rotating autoclave or in an autoclave equipped with mechanical agitators. A boiler on normal operation would undoubtedly provide comparable efficient mixing.

It is worthy of note that a number of the salts obtained by Ipatieff with hydrogen under pressure and temperature were similar to naturally-occurring minerals. Some of the deposits found in boilers have been likewise shown to closely resemble natural minerals.

It is also significant that the use of hydrogen under pressure has quite recently been adopted by base metal mining companies for the precipitation of copper, nickel, and cobalt after leaching with acid or ammoniacal solutions (1).

Evans, the eminent British authority on corrosion, states that hydrogen may be found in steam (2). It may be formed by the action of boiling water on iron, in the presence or absence of oxygen. Possibly part of the hydrogen comes from the decomposition of water by ferric hydroxide, giving magnetite, or it may be formed by the action of steam on hot steel at places which through faulty circulation have been starved of water. It should be recalled, too, that while the decomposition of water should theoretically give rise to chemically equivalent quantities of hydrogen and oxygen, the latter is removed in boiler systems by mechanical deaerators or reagents like sodium sulfite. The power plant operator is ever alert to the

* International Nickel Company of Canada, Ltd., New York.

danger of dissolved oxygen, whereas traces of hydrogen are assumed innocuous and allowed to remain in the boiler system.

We have then in boilers the presence of small quantities of hydrogen at the pressures and temperatures which Ipatieff and others have found could give the displacement of metals and metal oxides from solutions. We find occasionally in boilers certain deposits of metals, oxides, and salts which are identical with compounds formed by the action of hydrogen under conditions of pressure and temperature similar to those prevailing within the modern boiler.

Whether or not hydrogen plays any role in the origin of the deposits sometimes found in boiler drums and tubes is very difficult to determine, but it does appear possible that hydrogen could act in this manner.

REFERENCES

(1) Anonymous, "Hydrometallurgical Process Yields Pure Metal Powders from Sulphides," *Journal of Metals*, **5**, 775–9 (1953).

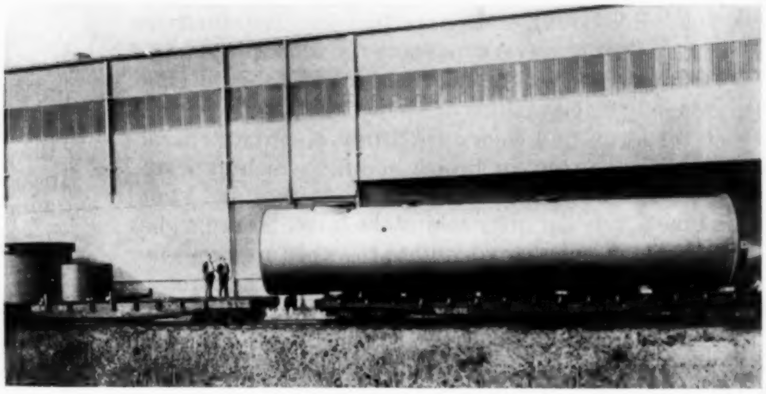
(2) Evans, U. R., *Metallic Corrosion, Passivity and Protection*, London, Edward Arnold & Co., 1946.

(3) Ipatieff, V. N., "Early Work on the Displacement of Metals and Metal Oxides," *Journal of Chem. Education*, **30**, 110–15 (1953).

(4) Young, R. S., and Hall, A. J., "Deposit Formations on Boiler Tubes," COMBUSTION, **20**, 41–44, Jan. 1949.

Utility Installs Giant Silencer on Gas Turbine Exhaust

What is claimed, by its manufacturer, Burgess-Manning Co., to be the largest silencer ever built, 74 ft long by 12 ft in diameter is pictured in the photo at right ready for shipment.



This particular gas turbine will be employed on a 5000-kw simple cycle gas turbine now operating in a midwestern public utility generating station. The gas turbine, particularly in the range up to 5000 kw, has enjoyed some fair acceptance by public utilities.

AIEE Winter General Meeting

LATEST advances and discoveries in electrical engineering were discussed and reviewed at the 1954 Winter General Meeting of the American Institute of Electrical Engineers held at the Hotel Statler, New York City, January 18–22. There were more than 400 papers and reports at the 95-session meeting for which about 5000 registered.

At the opening general session **Vice Admiral Harold G. Bowen,** Executive Director of the Thomas Alva Edison Foundation, commemorated the 75th anniversary of the invention of the first commercially successful incandescent lamp in an address entitled "The Genesis of the Light and Power Industry." He noted that the technology developed by this industry has been a vital factor in developing the security of the United States. Admiral Bowen urged that engineers take more interest in local education, particularly in elementary and secondary schools, in order to maintain the supply of engineers and scientists. He concluded with the statement that war will continue until all mankind is convinced that technology has made war unprofitable for all participants.

The Edison Medal which was instituted fifty years ago was presented to **John F. Peters,** engineering consultant to Westinghouse Electric Corp., for his work in fundamentals of transformer design and measurement of lightning voltages.

A. C. Monteith, vice president in charge of engineering, Westinghouse Electric Corp., was nominated as president of the Institute for 1954–1955.

The OVEC Project

Philip Sporn and **V. M. Marquis** of the American Gas & Electric Service Corp. enumerated some of the economic, engineering and financing problems of the 2,200,000-kw project of the Ohio Valley Electric Corp. (OVEC) to supply power for the gaseous diffusion plant near Portsmouth, Ohio. Two large steam-electric plants are being constructed by OVEC and its subsidiary, Indiana-Kentucky Electric Corp., one with a capacity of 1,200,000 kw and the other with 1,000,000 kw. Total expenditure for the two generating stations and the necessary 330,000-volt transmission lines is estimated to be $400,000,000.

In selecting plant sites the coal problem was recognized as basic from both economic and engineering viewpoints. It was considered desirable to tap at least two coal fields, those in western Kentucky-southern Indiana and in the eastern Ohio-Appalachian region. Both stations are located on the Ohio River, the larger known as Clifty Creek near Madison, Indiana, and the other Kyger Creek, near Gallipolis, Ohio.

It was decided to use indoor construction except for gas recovery equipment. The single boiler-turbine-generator arrangement was adopted, with each group of two units being segregated from the other units by a complete separation wall. Concrete for the sub-structure is being made of a mix wherein 20 to 25 per cent of the cement is being replaced by fly ash.

When in full operation the two stations are expected to consume 7,400,000 tons of coal per year, all of which is expected to be delivered by barge. Storage piles in excess of a million tons will be provided at each station.

The turbine-generators have a nominal rating of 200,000 kw, with throttle conditions of 2000 psig, 1050 F and reheat to 1050 F. The high-pressure machines operate at 3600 rpm and the low-pressure machines, at 1800 rpm. Average overall plant performance is expected to be under 9200 Btu per kwhr.

Furnace heat input per boiler is 1.87 billion Btu per hr, with a primary steam flow of 1,341,000 lb per hr and reheater flow of 1,190,000 lb per hr. Although both i-d and f-d fans are being furnished, the latter are sized for full pressurized furnace operation.

Combination mechanical and electrostatic precipitators will be used, with stack velocities of 120 fps. The stacks at Kyger Creek will rise 530 ft above grade line, while those at Clifty Creek will have a height of 680 ft. All stacks will be reinforced concrete, with two units per stack, except for one stack at Kyger Creek. With approximately 125 ft of height available in the base portions of the stacks, it was decided to locate the centralized control rooms in this region. Pneumatic-operated control systems will be used at Clifty Creek and electric-pneumatic control systems at Kyger Creek Station.

Generator Cooling

Sterling Beckwith and **L. T. Rosenberg** of Allis-Chalmers Mfg. Co. presented a paper entitled "A New Fully Supercharged Generator." They pointed out that the present completely supercharged generator is in many ways simpler and more rugged than the partially supercharged generator initially described two years ago. It takes full advantage of increased gas pressures and virtually has no load limit because of temperatures. Its axial core length is less than half that of the equivalent conventionally cooled machine, and its average and differential temperatures are lower, thus reducing the seriousness of differential expansion.

Through the more effective use of internal space the quantity of gas in this type generator has been cut in the ratio of 5 to 1 from the volume of its conventional predecessor. Its small bearings and seals minimize lubrication requirements, hydrogen consumption and mechanical maintenance. The weight and size reduction permits complete factory assembly of the largest machines and shipment without dismantling. It also affords the user substantial economies in power plant space and costs.

Turbine and Boiler Protection

In a paper entitled "Turbine and Boiler Protection and Interlocking on the A.G.&E. Co. System" **H. C. Barnes** and **C. P. Lugrin** of the American Gas & Electric Service Corp. based their remarks on a 200,000-kw cross-compound installation at the Kanawha River Plant of the Appalachian Electric Power Co. Two units are controlled from a single room located on the roof of the turbine room. Seven men are required for each shift, one watch engineer, one assistant watch engineer, two control operators, one assistant control operator, and two auxiliary equipment operators. There is no visual communication between the control room and equipment, and the men are responsible for the complete control of the boiler, turbine, condensate cycle, pump equipment, generator, auxiliary power switching and synchronizing control.

In order to insure availability of units it is necessary to give particular attention to interlocks to make sure that situations are not created which could cause accidental loss of an entire boiler or section because of control-circuit grounds, personnel error or unforeseen abnormal conditions. In the case of cross-compound turbines supplied by single boilers an outage of several hours may be required if it becomes necessary to remove field from the generators. The authors stressed the importance of maintenance of all equipment in a control system. They expressed the belief that a simple system involving some calculated risks and reliance upon operators is preferable to a

complicated one which tries to cover every contingency. This is based on the assumption that the simple system is understandable to the operators and is well maintained.

Gas-Turbine Starting

Three engineers from Westinghouse Electric Corp., **W. B. Boyum, R. W. Ferguson** and **J. G. Partlow,** presented a paper entitled "Methods of Starting Gas Turbine-Generator Units." In this paper, which will appear in more complete form in a forthcoming issue, they pointed out that it is necessary to accelerate a relatively high inertia load to approximately 60 per cent of rated speed in order to get a unit started. Where electrical power is not available, an internal combustion engine or an expansion gas turbine coupled to the unit through a clutch may be used. With available power a wound-rotor induction motor offers a satisfactory solution.

Research in Power

E. W. Boehne of the Electrical Engineering Department of Massachusetts Institute of Technology presented a paper entitled "An Application of Operations Research in the Power Field." Based on work sponsored by the Bonneville Power Administration, the report is a part of a long range program of economy studies of combined hydro-thermal systems.

Operations research may be used as a management guide (*a*) to indicate the optimum sequence of drawdown and refill characteristics for each reservoir over a short time and annual period, (*b*) to indicate the scheduling of a given quantity of irrigation water over a given annual period, (*c*) to indicate the optimum time to require that displacement steam should be added to the pool, (*d*) to indicate quantitatively the relative long term merits in terms of energy or cost of dropping specific loads for given periods, and (*e*) as a guide to the establishment of more profitable contractual agreements and rate structures.

Techniques of operations research also are useful as operating tools, as means for evaluating limitations, for project and multi-purpose planning, and as guides for flow forecasting, load estimating and load location.

"Active Research in the Development of Atomic Fuels for the Generation of Electric Power" was the title of a paper by **Walker L. Cisler** of The Detroit Edison Co. and **Alton P. Donnell** of the Vitro Corporation of America. They pointed out that re-

search in the field of nuclear power involves the development of completely new workable methods for applying this new science and heat resource, so that it will serve a useful purpose and so that its use can be economically justified. The new methods must be drastically lower in cost than those now available if the real energy potential is to be realized.

Commenting upon the coordination of research efforts, the authors stated that there is now a substantial measure of cooperation among the electric power operating industry, the equipment manufacturing industry and the Federal Government. These efforts are not fully cordinated, it was noted, because of the present Atomic Energy Act which in effect provides for a government monopoly of atomic energy. This law does not allow industry to spend large sums of private funds because there is no assurance that the investment could ever be owned or recovered.

Discussing "Research for and by the Electric Power Industry," **J. E. Hobson** of Stanford Research Institute and **W. A. Lewis** of Illinois Institute of Technology urged that basic research of a long-term nature be undertaken. This should not replace but should supplement and complement that done by manufacturers and suppliers of electrical equipment. It was proposed that the annual industry-wide budget should be about 0.1 per cent of gross electric utility income, or $5,500,000.

Several other recommendations concerned the balancing of research efforts and public relations aspects of research. It was suggested that every effort should be made to utilize this program to improve the scientific manpower situation by (1) providing stimulating problems within the industry, (2) encouraging college and university faculty members to study the industry's problems, and (3) demonstrating to students that the industry has interesting work to do.

Cooling Generators

"Direct Cooling of Turbine-Generator Field Windings" was the title of a paper by **C. H. Holley** and **H. D. Taylor** of General Electric Co. The heat from the I^2R loss is removed directly by cooling gas flowing through passages inside the winding conductors, thus eliminating the thermal drop in the ground insulation. This is known as direct cooling. A further characteristic of the system used by the authors' company is that much of the rotor cooling gas is picked up by scoops from the air gap between the

stator and rotor, this being designated as an air-gap pickup type of cooling.

For a first application it was decided to try this construction on a fairly large generator (on the order of 125,000 kva at 30 psig). Benefits were expected to show up as a major reduction in the temperature rise of the field winding, no attempt being made to reduce the physical size of the machine. The generator was completed in 1953 and was subjected to comprehensive factory tests before being shipped to the Huntley Station of Niagara Mohawk Power Corp.

The tests disclosed that the new construction did not appreciably impair the efficiency of the turbine-generator unit. The rotor met all expectations electrically and operation was very smooth without appreciable difference in noise level. When operated with hydrogen at 30 psig the new field had a temperature rise of about 25 per cent that of the standard field.

R. A. Baudry and **P. R. Heller** of Westinghouse Electric Corp. were the authors of a paper entitled "Ventilation of Inner-Cooled Generators." In conventional units all the coil losses must be dissipated through the electrical insulation so that inherently an appreciable thermal gradient occurs at this point. To remove the limitation it was necessary to develop a more effective cooling means by which the thermal barrier of the electrical insulation could be eliminated. This could be achieved by circulating the cooling in direct or nearly direct contact with the conductors.

The coolant is introduced in a manner so that the centrifugal forces generated by the rotor are employed to generate part of the differential pressure producing flow in the coils. In the determination of coil rating the limitation is imposed by the maximum temperature at which the conductor may operate in contact with the electrical insulation.

In concluding the authors pointed out that this method of cooling rotor and stator windings makes it possible to build machines of high specific ratings with moderate hydrogen pressures. The reduction in the size of the rotor plus the reduction in the quantity of gas circulated through the machine permits the use of high-pressure blowers and results in windage losses lower than on conventional units. Maximum copper temperatures are the same as obtained on the standard windings, while the average temperatures have been reduced, which is the desired objective.

American Power Conference Program

THE sixteenth annual meeting of the American Power Conference will be held on March 24, 25 and 26, 1954, at the Sherman Hotel in Chicago. The Conference is sponsored by the Illinois Institute of Technology in cooperation with twelve universities and ten local and national engineering societies. The purpose of the conference is to provide a forum for the exchange of information in the fields of power generation, transmission, distribution and utilization. The major portion of the preliminary program is contained in the following listing.

Wednesday, March 24, 9:00 a.m. Registration

10:00 a.m.–12:00 Noon. Opening Meeting

"Research and the Electric Power Industry," by J. E. Hobson, Stanford Research Institute, and William A. Lewis, Illinois Institute of Technology, Chicago.

12:15 p.m. American Power Conference Luncheon

"The Power Industry—A Challenge to Engineers," by Walter H. Sammis, president of Ohio Edison Co. and Edison Electric Institute.

2:00–5:00 p.m. Central Station Steam Generation

"Performance of New Controlled Circulation Boilers," by E. M. Powell, Combustion Engineering, Inc.

"Cyclone-Furnace-Fired Boilers," by George W. Kessler, The Babcock and Wilcox Co.

"Pulsation-Induced Vibration in Utility Steam Generation Units", by Raymond C. Baird, The Fluor Corporation, Ltd.

"The Present and Future Status of the Fly Ash Disposal Problem," by C. M. Weinheimer, The Detroit Edison Co.

2:00–5:00 p.m. Hydroelectric Power Development in the United States

"The Role of the Corps of Engineers in the Development of Water Resources," by Brig. Gen. E. C. Itschner, C. H. Giroux, Gail A. Hathaway and Col. G. J. Zimmerman, Office of the Chief of Engineers.

"The Federal Multiple-Purpose Project—Its Role in the West," by H. B. Taliaferro and W. L. Newmeyer, Bureau of Reclamation.

2:00–5:00 p.m. Industrial Plant Session

"Scale Modeling—A Practical Engineering and Construction Tool," by James A. Carroll, The Procter and Gamble Company.

"Selection, Maintenance, and Piping Practice in Industrial Plants," by Robert J. Pinske, Crane Company.

"Planning and Installing an Electrical System in a Rapidly Growing Industrial Plant," by Hans Ederegger, Jr., I. L. Weldy and Associates, and M. W. Stehr, Wisconsin Electric Power Co.

3:30–5:00 p.m. Industrial Electrical Session

"Recent Developments in Pipeline Electrification," by Merritt Hyde, Westinghouse Electric Corp.

"Open Hearth Versus Electric Furnace Economics and Their Significance to the Power Industry," by David D. Moore, Battelle Institute.

Thursday, March 25, 1954, 9:00 a.m.–12:00 Noon. Central Station Steam Turbines

"Steam Turbine Development," by Clarence C. Franck, Steam Division, Westinghouse Electric Corp.

"Trends in Design of Present Day Steam Turbines," by Charles D. Wilson and Ellis P. Hansen, Steam Turbine Department, Allis-Chalmers Mfg. Co.

"The Steam Turbine of Tomorrow," by R. S. Neblett, Turbine Division, General Electric Co.

9:00 a.m.–12:00 Noon. Fuel Economics

"Economic Trends in the Use of Natural Gas," by Richard Gonzales, Humble Oil Co.

"Future of Coal in Power Generation," by George A. Lamb, Pittsburgh Consolidation Coal Co.

"Nuclear Fuels for Power Generation," by Walter F. Friend, Ebasco Services, Inc.

10:30 a.m.–12:00 Noon. Water Technology

"Evaluation of Several Alkaline Compounds for Controlling Corrosion in Boiler Feedwater Systems," by J. M. Decker and J. C. Marsh, The Detroit Edison Co.

"High Temperature Water for Process Heating Combined with Power Production," by Paul L. Geiringer and Floyd Hasselriis, American Hydrotherm Corp.

2:00–5:00 p.m. Central Station Steam Power Plants

"Supercritical Pressure Steam Power Cycles," by Prof. Jerome Bartels, Polytechnic Institute of Brooklyn, N. Y.

"The Gallatin Steam Plant of the Tennessee Valley Authority," by C. E. Blee and H. J. Petersen, Tennessee Valley Authority.

"The Economy of Large Generating Units," by Howard P. Seelye and William W. Brown, The Detroit Edison Co.

2:00–5:00 p.m. Water Technology

"Demineralized Water for 1500-psi Steam Plant—Design Aspects," by C. R. Stewart, Stone and Webster Engineering Corp.

"Demineralized Water for 1500-psi Steam Plant—Operating Aspects," by W. B. Gurney, Gulf States Utilities Co.

"Automatic Mixed Bed Demineralizing at the Niagara Mohawk Power Company," by Durando Miller, The Permutit Co., and T. J. Finnegan, Niagara Mohawk Power Co.

"The Expected Life of Anion Exchangers Under Various Conditions of Deionizer Design and Operation," by Louis Wirth, National Aluminate Corp.

2:00–3:30 p.m. Industrial Power Plants—Economic Aspects

"Economic Factors Affecting Selection and Replacement of Power Plant Equipment," by Gerald J. Matchett, Illinois Institute of Technology.

"Hawthorne Power Plant Rehabilitation Economics," By C. E. Morrow and R. F. Born, Western Electric Co.

2:00–3:00 p.m. Electrical Systems

"Load Structure of a Modern Electric Utility System," by Constantine W. Bary, Philadelphia Electric Co.

"Service Standards in Relation to Reserve and System Design," by W. J. Lyman and V. E. Hill, Duquesne Light Co.

6:45 p.m. All-Engineers Dinner

Speaker: Douglas McKay, Secretary of the Interior.

Friday, March 26, 1954, 9:00 a.m.–12:00 Noon. Nuclear Energy

"Economic Aspects of Various Types of Nuclear Reactors," by Donald H. Loughridge, Northwestern Technological Institute.

"Technology of the Use of High-Pressure Water for Reactors," by A. Amorosi, Reactor Engineering Division, Argonne National Laboratory.

"Problems of Operation of Nuclear Power Plants," by R. L. Doan, Atomic Energy Division, Phillips Petroleum Co., National Reactor Testing Station.

9:00—10:30 a.m. Water Technology

"Water Problems in the Nuclear Power Field," by R. C. Ulmer, Combustion Engineering, Inc.

"Boilers and Boiler Waters—Interlocking Advances in Design," by H. M.

Rivers and S. R. Osborne, Hall Laboratories, Inc.

9:00—10:30 a.m. Steam and Diesel Power Plants

"Steam Versus Diesel Power Plants," by C. M. Stanley, Stanley Engineering Co.

10:30 a.m.–12:00 Noon. Water Technology

"Silica Removal by Salt Splitting without Demineralizing," by S. B. Applebaum, Cochrane Corp., and B. W. Dickerson, Hercules Powder Co.

"Some Chemical Aspects of Hot Process-Hot Zeolite Plant Performance," by M. Lane and J. H. Duff, Graver Water Conditioning Co.

10:30 a.m.–12:00 Noon. Gas Turbines for Power Generation

"Gas Turbines for the Power Industry," by T. L. Putz, Westinghouse Electric Corp.

"A New Power Cycle Combines Gas Turbine with Steam Turbines," by Louis S. Gee, West Texas Utilities Co.

12:15 p.m. American Power Conference Luncheon

Speaker: Thomas E. Murray, Atomic Energy Commission.

2:00–5:00 p.m. Developments in Gas Turbines and Diesel Engines

"Performance of 2400 Hp *Trainmaster* Diesel Locomotives," by Robert Aldak, Fairbanks Morse and Co.

"Use of Gas-Turbine-Electric Locomotives on the Union Pacific Railroad Company," by F. Fahland, Union Pacific Railroad Co.

"Factors Associated with Use of Gas Turbines for Automotive Applications," by John H. Bonin and Robert A. Harmon, Armour Research Foundation.

"Gas Turbines in the Steel Industry," by George H. Krapf, United States Steel Corp.

2:00–5:00 p.m. Industrial Steam Generation

"Some Economic Factors Influencing Industrial Boiler Manufacture," by Carl E. Miller, Combustion Engineering Inc.

"Operating Experiences with a Multi-Fuel Stoker-Fired Boiler," by G. G. Bachman, Omaha Public Power District.

"Industrial Operating Experience with Cyclone-Fired Boilers," by Leo L. Moran, The Dow Chemical Co.

Editor's Note: The complete and final program, when available, may be secured from Professor R. A. Budenholzer, Director, American Power Conference, Illinois Institute of Technology, Chicago 16, Ill.

REVIEW OF NEW BOOKS

Any of the books here reviewed may be secured through Combustion Publishing Company, Inc., 200 Madison Ave., N.Y.

Theory and Design of Steam and Gas Turbines

By John F. Lee

There is a pedagogical need for a text in which the fundamental theory of steam and gas turbines is presented in a unified form in a single book. It is the intention of the author, who is Associate Professor of Mechanical Engineering at North Carolina State College, to fulfill this requirement. As a consequence of this attempt to discuss two types of prime movers there is an inherent shifting back and forth from one to the other.

Following a presentation of basic steam and gas-turbine types, the author has included a concise review of thermodynamics. Steam and gas-turbine cycles are then taken up, after which chapters are devoted to gas dynamics, nozzle design and energy interchanges in fluid machinery. Common aspects of steam and gas-turbine design are discussed in chapters on flow passages and mechanical problems. Consideration of the steam turbine is completed with a chapter on its control and performance. The remainder of the text is concerned with centrifugal and axial-flow compressors, problems of combustion, the regenerator and the gas-turbine power plant.

There are 502 pages in the text which has a selling price of $9.

The Atomic Submarine and Admiral Rickover

By Clay Blair, Jr.

Published three days before the launching of the atomic-powered submarine *Nautilus*, this is the dramatic story of the man who is considered most responsible for the promotion of the U. S. Navy's nuclear energy projects. Most books reviewed in these columns are objective engineering studies which are devoid of controversy. That is not true in the case of this book, which is the account of a very nonconformist officer, his struggle for promotion to admiral, and his development of a team which was responsible for the design of the *Nautilus*.

In the opening of the book mention is made of some of the early Navy work on reactors at Oak Ridge in which Captain Rickover and others had a part. Interspersed with this is an account of his career from Naval Academy days to the inception of the atomic-powered submarine project.

A second section of the book is concerned with the design stages prior to building the *Nautilus*, with much emphasis upon internal struggles and some of the industrial relationships. A succeeding section titled "The Admirals Intervene" deals with the campaign for the promotion of Captain Rickover.

The author is a correspondent for *Time* and *Life* magazines and writes very much in the style of those publications in championing Admiral Rickover's cause. He has prepared an exceptionally readable book that should receive a great deal of attention.

The 278-page book is priced at $3.50.

Betz Handbook of Industrial Water Conditioning, Fourth Edition

First issued in 1942, this informative and well-illustrated handbook is now in its fourth edition. The introductory chapters of the handbook deal with such basic water-treatment processes as aeration, coagulation, filtration, chlorination and softening. The following chapters are concerned with specific water problems encountered in the fields of boiler-water conditioning and treatment of cooling-tower waters.

A supplementary section of the handbook is devoted to water control analyses and their interpretation. Methods of testing are provided for turbidity, hardness, calcium alkalinity, free carbon dioxide, sulfate, chloride, silica, phosphate, sulfite, dissolved oxygen, hydrogen sulfide, iron, manganese, chlorine, nitrate, chromate and ammonia.

The handbook has been carefully organized for quick reference or self-study. It should be useful both to those with considerable experience in water conditions and those who desire to become acquainted with current practice.

Printed on coated stock, $8^1/_4 \times 10\ ^3/_4$ in., with flexible cover, the 248-page book sells for $3.

Pilot Plant Catalytic Gasification of Hydrocarbons

By C. H. Riesz, P. C. Lurie, C. L. Tsaros and E. S. Pettyjohn

This report sponsored by the Gas Production Research Committee of the American Gas Association presents results of a pilot plant investigation of the catalytic cracking of hydrocarbons of low molecular weight, in the presence of steam and air, as a method of producing equivalents of various types of manufactured and natural utility gases. The work was conducted by the Institute of Gas Technology, Chicago, with the cooperation and assistance of the Philadelphia Electric Co. at the latter's Chester, Pa., station.

Hydrocarbons gasified in this study included natural gas, refinery oil gas, propane, butane, gasoline, kerosene, crude naphtha and light gas oil. It is shown that cracking them to produce a low heating value "carrier" gas, and subsequent enrichment of this carrier gas with natural gas or propane, is a means of providing base load substitutes or peak load supplements for utility systems distributing carbureted water gas, coke oven-carbureted water gas mixtures, natural gas or manufactured gas-natural gas mixtures.

There are 44 pages in the paper-bound report which sells for $5.

Procedures in Experimental Metallurgy

By A. U. Seybolt and J. E. Burke

Metals and their alloys are matters of considerable interest to power plant technicians and especially so to design and specifications personnel. This book has been developed as a guide in establishing laboratory techniques for the preparation of metals and alloy specimens for further study.

The authors are well qualified for the subject at hand. Dr. Seybolt is research associate with the metallurgy research department in General Electric Co.'s Research Laboratory and Dr. Burke serves as manager, the metallurgy section, of Knolls Atomic Power Labs.

The guiding principle for the book is the preparation of metal samples up to the point of making observations on the properties of the metal. Accordingly the authors have elected to approach their main points by way of several preliminary chapters on high temperature covering how it is obtained, how measured and controlled and what refractories are available for high temperature service. In addition the vital problems of controlled atmospheres and vacuum developing systems are treated in separate chapters.

Over and above descriptions of different laboratory means of producing a sample of a desired structure, chapters are included on the specialized subjects of powder metallurgy and the fabrication of metals.

The book, cloth bound, 6 in. by $9^1/_4$ in., contains 340 pages and sells for $7.

New Catalogs and Bulletins

Any of these may be secured by writing Combustion Publishing Company, 200 Madison Avenue, New York 16, N. Y.

Johns Manville Catalog

Essential data on insulation, refractory products, asbestos cement pipe, packings, gaskets, electrical products, frictional material plus roofing, siding, and other building construction supplies form the subject matter of the new 40-page Industrial Products Catalog of the Johns-Manville organization.

Feedwater Deoxygenation

The successful use of hydrazine for the removal of trace oxygen from boiler feedwaters is told in a 16-page bulletin supplied by the Mathieson Chemical Corp. Application data including drawings, method of use, handling precautions are furnished. In addition reprints from two national magazine articles are included to give material of interest to the power and petroleum fields.

Postweld Heat Treatment

Increased use of austenitic steels for piping and pressure vessel fabrication has given rise to considerable confusion and misunderstanding on the need for stress relief and the effect of this heat treatment on weldments already in service. The American Welding Society has summarized in a booklet, "Recommended Practices for Postweld Heat Treatment on Austenitic Weldments," all the necessary considerations to determine whether stress relief is applicable in a given case and if so what temperature and time is best.

Deaerating Heater

A specially designed deaerating heater for small and medium size power plants, the Uni-Pac, features Cochrane Co. publication No. 4643. Operation, design, capacities and advantages are described.

Thermometer Bulbs

The Industrial Div. of Minneapolis-Honeywell Regulator Co. in its recently released bulletin, 5710, describes resistance thermometers of high speed, marine room and sanitary types for temperature spans as narrow as 20 F. Wet and dry bulb assemblies for relative humidity measurement are also included.

Personals

L. A. Kilgore, formerly assistant manager of generator engineering for the Westinghouse Electric Corp.'s transportation and generator division has been made staff engineering manager for the East Pittsburgh divisions and will direct the engineering laboratories at East Pittsburgh.

Charles B. Sweatt has been elected to the newly created post of vice-chairman of the board of Minneapolis-Honeywell Regulator Co. **Tom McDonald** and **A. M. Wilson** have been made executive vice presidents in the same organization to assist **Paul B. Wishart,** president.

Peer A. Abetti, advance and development engineer, with the Pittsfield, Mass., works of General Electric Co. has been selected the nation's outstanding young electrical engineer for 1953 and granted the Eta Kappa Nu award for his "original approach to power transformer design through the creation of unique electromagnetic models and his exceptional civic and cultural attainments."

Reliance Electric & Engineering Co. recently promoted **C. V. Gregory,** Pittsburgh district manager, to the post of manager of district sales. **E. H. Koontz** of the Newark, N. J., sales office succeeds Gregory in Pittsburgh and **F. Raymond Obenchain** takes over Koontz's duties. **C. B. Allen, Jr.** succeeds recently retired **J. L. Buell, Jr.** as Detroit district manager.

Harry A. Winne, retired recently by General Electric Co., has been elected to fill a vacancy on the board of directors of the American Gas and Electric Co. In addition three newly created executive vice presidents—**Graham Clayton, H. A. Krammer** and **Donald C. Cook**—were moved up from their former positions as vice presidents.

Westinghouse Electric Corp. has announced the appointment of **Latham E. Osborne,** formerly executive vice president of defense products, to the position of executive vice president and also his election to membership on the company's board of directors. **Leslie E. Lynde,** vice president of the aviation gas turbine div., succeeds to the defense products post in which assignment he will be responsible for the atomic power division among others.

George E. Best, Mutual Chemical Co. of America, has been elected to a three-year term on the National Association of Corrosion Engineers' board of directors to succeed **L. B. Donovan** of Consolidated Edison Co. of N. Y.